The
ENGINEERS' DIGEST
REVIEW OF WORLD ENGINEERING PROGRESS
VOL. XIV No. 6 JUNE, 1953

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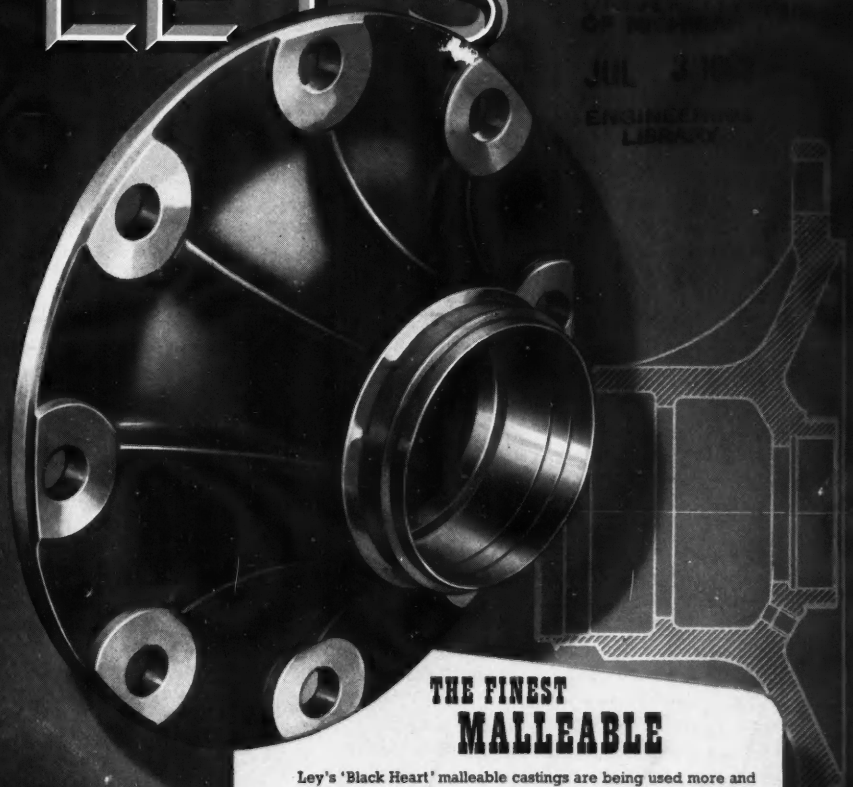

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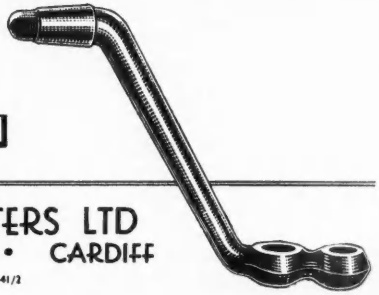

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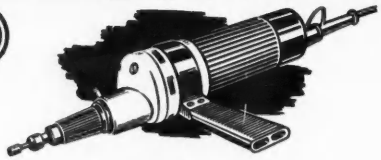

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REVIEW OF WORLD ENGINEERING PROGRESS

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THE BLUE PAGES

A QUICK GLANCE AT THE LATEST TECHNICAL DEVELOPMENTS THROUGHOUT THE WORLD

● *NEW TYPE OF GASKET*

A new type of gasket, which is claimed to provide an improved seal at lower labour and material cost, has recently been introduced as a replacement for conventional pre-cut cork or rubber gaskets. Termed the " flow-in " gasket, it is applied as a liquid synthetic rubber or resin compound forced through a nozzle onto a spinning component part and then baked to form a solid rubbery gasket which will not fall off the part. With this method the gasket can be applied either automatically or semi-automatically, depending upon the lining equipment used, and in varying types, thicknesses and diameters. The manufacturers claim that the chief advantage over conventional pre-cut gaskets is lower cost of material and faster application, since the gasket does not have to be crimped, stamped or glued to hold it in place. On the automatic lining machine, which can apply gaskets at speeds up to 300 parts per minute, depending upon the size of parts to be lined, the parts are belt or gravity fed to the machine and then to the chuck. The chuck spins the part under an adjustable nozzle, which squirts a measured amount of compound into a groove or channel on the part. The lined part then moves off onto a conveyor belt to an oven where it is baked. The machine can handle parts from 3½ to 12 in. in diameter. If belt-fed, high speed equipment is used and no operator is required. In the case of a semi-automatic machine, the part is placed on the chuck by hand. A special advantage of the new method appears to be the possibility of using specially formulated compounds to give effective seals against moisture, oils, petrol, vacuum, vibration, etc., as the case may be.

● *MEASURING CASE DEPTH*

Among the latest applications of radioactive substances to the investigation of metallurgical problems is the measurement of the case-depth, as represented by the depth of carbon absorption in a case-hardened steel. For this purpose, the two radioactive isotopes carbon-14 and cobalt-60 are being employed with success. These two isotopes are for all practical purposes chemically identical with the normally occurring isotopes carbon-12 and cobalt-59 respectively, except that carbon-14 and cobalt-60 are radioactive. By adding a minute quantity of radioactive carbon to the stable isotope, a homogeneous mixture is obtained and the radioactive isotope acts as a tracer or visible indicator of the behaviour of the carbon. Detection of the radioactivity with a Geiger counter will indicate that carburisation has taken place, but will not give an indication of the depth of carbon absorption affected. A very simple method for determining the depth of carbon absorption, is, however, available. A sufficient amount of radioactive carbon is introduced in the carbon to be used in carburising the steel. After carburising, sections of the steel are put into direct contact with X-ray film. After exposure, the films are developed and the pictures obtained reveal the pattern of the carbon distribution in the carburised steel. It is, of course, necessary to effect close direct contact between film and the section studied. Also, sufficient radioactivity must be present in the carburised layer in order to avoid long exposures in obtaining a photographic print. An analagous method could be employed to study the effect of fusion welding upon carbon content in the weld and in the parent metal.

● *BRAZING AND SOLDERING ZIRCONIUM*

Zirconium and its alloys cannot be soldered, brazed or hot-dip coated by conventional methods such as used for treating other metals. Ordinary fluxes either do not remove the zirconium surface film, or do not provide a surface easily wet by the brazing or hot-dipping alloy. It has been found that zirconium parts can be soldered and brazed by regular methods if they are dipped in molten zinc chloride beforehand. This pre-treatment deposits a thin coating of zinc on the surface. Hot-dipped coatings of tin or lead may also be applied after the zinc dip. The desired zinc film is deposited on zirconium from molten zinc chloride in the temperature range of 825° F to 840° F. The need for the sublimation of the zirconium tetrachloride formed at the zirconium surface established the minimum temperature of zinc coating at about 570° F. However, temperatures above the melting point of zinc (786° F) give the best zinc films. After the dipping, the zirconium part is removed, drained and cooled in air for about 15 seconds and then quenched in water to wash off the entrained salt. A smooth matt coating of zinc remains on the zirconium surface. Parts can then be tinned, soft-soldered or silver soldered by conventional means. The

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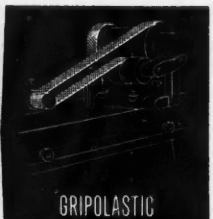
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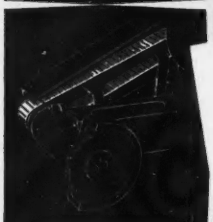
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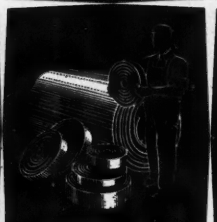
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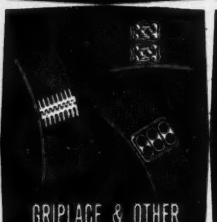
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strength of the bond between the zirconium and the brazing or hot-dip coating was evaluated by qualitative tests. In the case of a soft soldered piece, the strength of the joint was found greater than the strength of the soft solder, while in the case of a silver soldered piece, the joint approached the strength of the silver solder. Attempts to coat titanium by immersion in molten zinc chloride proved unseccessful.

● *TESTING SHEET STEEL FOR DRAWABILITY*

It is reported that a simple method has been found for determining the drawing qualities of flat rolled sheet steel. With this method, which supersedes the corner bend test, pre-testing of steel sheets can be done on the shop floor immediately before they are used for blanking or stamping. The new test is said to supplement the more involved cup and tensile tests. Owing to the ease and speed of testing, it is possible to make a quick check of sheet steel drawing quality when a sudden change of performance occurs in the draw die. Also, a shipment of steel can be readily sorted into high, medium, and low drawing quality material. Moreover, it is expected that the new testing device will throw more light on the need for aluminium-killed steel. It should also help to establish reliable quality control standards for stamping jobs and eliminate costly stamping of unsuitable steel. The principle of the new test is similar to that used by experienced die and press men when they bend the corner of a sheet of steel to " feel " its resistance. The new test device accurately measures the resistance of steel to a controlled bending force and determines the degree of curvature of the bend made by that force. The test is a quantitative measure of probable success or failure of sheet steel during the drawing operation. The resistance of this sheet to the bending force is recorded on a dial gauge, yielding an indication in arbitrarily established units which must be corrected in order to allow for variation in sheet thickness. The maximum curvature is measured by a spherometer.

● *NEW PLASTIC FOR WIRE INSULATION*

It would appear that some of the most interesting insulating materials of recent development are the polymers of monochlorotrifluoroethylene, known as fluorothene. These materials have extremely interesting properties, outstanding among which is an excellent chemical resistance combined with thermal stability at the relatively high temperatures of 125-150° C. Owing to the completely halogenated structure of its molecule, the material exhibits extremely good resistance to burning. Tests on insulated wire of various sizes and wall thicknesses have shown that the material will not support combustion, in other words it will not continue burning after application of a flame. While many insulating materials have excellent resistivities at room temperature or slightly above, most of these materials have the undesirable characteristic of dropping sharply in resistivity as the temperature is raised. Fluorothene, on the other hand, has volume resistivity values beyond the upper range of equipment generally used for these measurements at ordinary temperatures. The surface resistivity of fluorothene is also extremely high, as the surface of this resin has the unique property of being very difficult to wet with water. Therefore, conducted paths depending on the presence of electrolytes or moisture are difficult to develop. Both the dielectric constant and the power factor are excellent for use over the normal range of frequencies. At the higher frequencies, however, the dielectric losses will generally prohibit the use of fluorothene. The chemical resistance of fluorothene is particularly outstanding, as there is no known solvent for the polymer in the operating range contemplated. It is particularly resistant to the effects of fuming nitric acid.

● *METAL POWDER FILTERS*

The constant demand for increased purity of liquids and gases in industrial practice has greatly increased the employment of filters of all types in all branches of industry. In many applications the filter material is required to do more than mere filtration. Thus, for instance, the filter may have to serve for diffusing streams of liquid and gases, for separating liquids having different surface tensions, or for metering or controlling the flow of gases and liquids. In another important application, filter material is used as a flame barrier on equipment employed in or near explosive gases. Porous metal filters prove very useful also as sound arresters in compressed air operated equipment, such as pneumatic hammers, where the exhaust air is passed through powdered metal filters placed in the escape openings. These various requirements are particularly well met by filters of the well-known sintered metal powder type, which may be made of several metals, e.g., stainless steel, nickel-silver, etc., in addition to the more familiar bronze. Unlike ordinary screen filters, sintered metal powder filters are capable of depth filtration. The fluid being filtered has available innumerable paths through which it may flow; if deposited foreign particles block direct passage, the fluid is free to pass in other directions until it finds escape openings. It is thus not confined to flow along one level or only in one direction. Porosity of sintered metal powder filters is of paramount importance and can be accurately controlled from almost zero to some 50 per cent. A coating can be applied to protect the surface of the filter if the nature of the fluid makes it desirable; thus, in the case of petrol filters, tin plating may be used to prevent gumming. Corrosion-proof filters made of stainless steel powder are also available.

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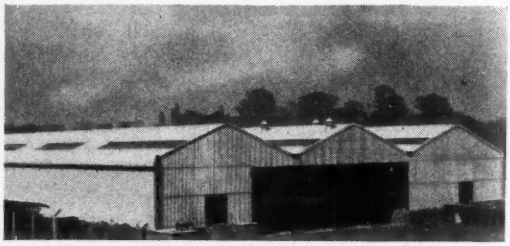

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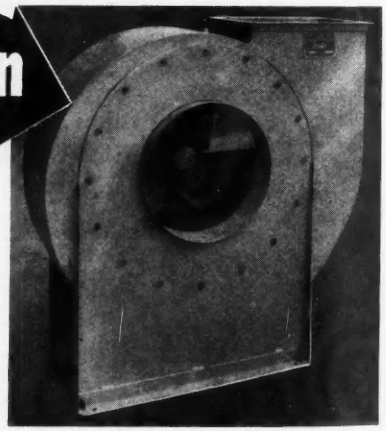

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● *LIGHTWEIGHT CARRIER CABLE*

It is reported that a new type of carrier cable developed by the U.S. Army Signal Corps has ten times the impact resistance, five times the frequency range, weighs about one-third less, and yet costs about one-third less than a corresponding type used in the last War. This cable is of the multi-channel carrier communications type, and its insulation is made of polyethylene, while its outer jacketing is made of a new Vinylite plastic. The Vinylite plastic is capable of withstanding extremes of heat, cold, pressure flexing, and chemical corrosion. The maximum operating range for this Vinylite plastic extends from 140° F to minus 67° F, and it is claimed that the material maintains excellent abrasion resistance, hardness and toughness throughout this temperature range. The centre of the new cable consists of four conductors of seven-strand copper wire, each covered with extruded polyethylene insulation. The four conductors are wound in a spiral and held firmly together by a belt of extruded polyethylene. Over this belt, carbon-impregnated textile tape is wrapped for electrostatic stabilisation. Over this tape is laid a stainless steel braid to furnish high tensile strength and yet avoid the build-up of electrical losses at 60 kilocycles. Finally, a jacket of Vinylite plastic is extruded over the steel braid. The frequency range of the cable includes twelve channels to 60 kilocycles and there is a separate supervisory voice frequency between zero and four kilocycles, used only for maintenance.

● *PREVENTING STAINLESS STEEL GALLING*

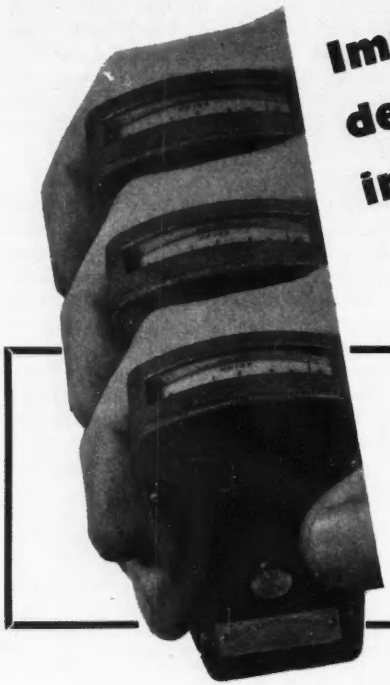
Where ordinary lubrication methods are not applicable, the prevention of galling and seizure of stainless steel surfaces in mutual rubbing contact presents a considerable problem. The results of an investigation of the galling characteristics of various stainless steels will, therefore, prove valuable. A special galling test was developed for this purpose after a number of methods involving the use of torque wrenches, strain gauges and nut and bolt assemblies had been studied. In the new test, the polished base of a cylinder section is rotated under pressure against a polished block surface for one revolution, a series of such assemblies being compressed under increasing loads until galling occurs. It was found from these tests that stainless steel sections at a relatively high hardness level or with a substantial difference in hardness generally exhibit better resistance to galling than the combination of two soft members. Since there may be no change in chemical composition, this relationship between hardness and resistance to galling is probably due to the difference in mechanical properties of the hardened and of the soft specimen. The explanation has been put forward that the hardened sections deform elastically near the contact points under loading, while the softer, weaker pieces yield plastically for a significant distance beneath the contact points. During movement, hardened surfaces may recover elastically with decrease in pressure, and this motion tends to sever any metallic welding, whereas, in the case of the soft specimens, the surfaces probably continue to adhere closely during the movement of the specimens with relatively little elastic recovery and separation.

● *X-RAY GAUGING*

The automatic gauging of thickness without physically contacting the material being measured may find many useful applications in machine tool control. One of the latest methods is X-ray gauging, which can be applied to any material which absorbs X-rays, and which has already been employed with success for metal gauging and in connection with rubber processing machinery. In the first case, an automatic gauging machine of this type has effected a reduction in the time required for gauging extruded propeller blading from 8 hours to less than 1 hour. The survey was formerly performed by hand with mechanical calipers, 18 ft in length. Essentially, the method constitutes a comparison process in which the X-ray penetration through the propeller blade is compared with the penetration of an X-ray beam of equal strength through a metal sample of known thickness. For this purpose, two X-ray beams approximately $\frac{1}{16}$ in. in diameter pass through the standard sample and the propeller blade respectively. Cadmium sulphide crystal detectors, placed inside the propeller blade and next to the reference sample respectively, are employed to detect the variation in X-ray intensity. The amplified detector outputs are then applied to a bridge circuit to produce a deviation reading. In order to ensure high accuracy, the output of the reference sample detector is measured by an error amplifier which, in turn, operates a control which, by adjusting the X-ray tube input, holds the reference output at a fixed value. The blade detector is supported by a long, thin shaft, which allows it to pass into the narrow portions of the blade.

● *ELECTROLYTIC TANK DEVELOPMENT*

A recent report describes the design and development of an electrolytic tank for the study of hydrodynamic flow about torpedoes and other bodies. In this tank it is possible to survey velocity distributions of the potential flow about two-dimensional and three-dimensional bodies of revolution with an accuracy comparable to that obtained by direct measurement in the towing basin or water tunnel. The method may eventually be extended to study more general types of flow on three-dimensional bodies, e.g., pressure distributions on a ship model. Electrolytic tanks, have, of course, been built at many laboratories in various countries, but there still remains much scope in the refining of testing techniques as, for



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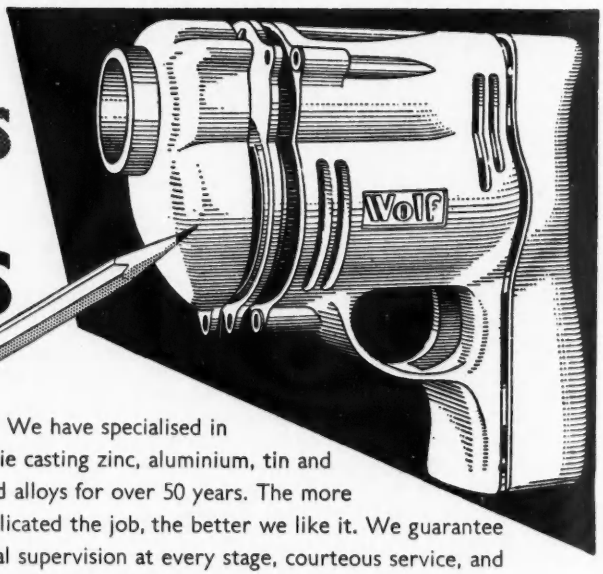

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instance, for obtaining pressure distributions. Also, considerable care is necessary in designing and constructing an electrolytic tank. The lining of the tank described in the report was cast of a wax compound and the surfaces were made smooth and plane by hand scraping. This type of lining was adopted after considerable trouble was encountered with a glass lining. The dielectric models investigated were machined out of the same hard wax compound used for ship models, and sprayed with a hydrophobic waterproofing plastic compound to eliminate the meniscus of the water surface. Polarisation of the electrolyte adversely influences accuracy of readings, but polarisation may be reduced by increasing the frequency of the power supply and by coating the probe and electrodes with Aquadag.

● *NEW TYPE SPECTOGRAPH*

A new type of spectograph has been perfected which, owing to its greater accuracy, should prove a useful instrument both in research and in industrial practice. It is claimed that because of the combination of a prism and an echelle, the instrument makes it possible to carry out quick routine analyses in the proportion of one part in a million or less. An echelle is described as a new optical element conceived a few years ago. Before the echelle was developed, spectographs were being built larger and larger so as to obtain maximum utilisation of prisms and diffraction gratings. However, space limitation made it difficult further to increase the size of spectographs; the echelle offers a new solution. As used in the new instrument, an echelle is an inch-thick optically flat glass plate of 3×5 inches in which grooves have been ground accurately, 200 to the inch. Like a prism or a diffraction grating, an echelle disperses light into a spectrum so that the composition of the light-emitting source can be studied in detail. The dispersive power of an echelle, however, can be ten or more times that of a diffraction grating. Its resolving power or ability to show the separation of spectral lines is doubled or trebled. In the new instrument, the echelle is used in conjunction with a prism, so that the prism spreads out the spectrum horizontally on the spectographic plate, and the echelle spreads it out vertically. This produces a two-dimensional spectrogram (called an "echellogram") of greatly increased value in identifying and measuring the substance under study.

● *COLD REDUCTION OF STEEL TUBING*

The production of a steel tube by reduction of a tube of larger diameter offers several important advantages as far as a finished product itself is concerned. First of all, the tube reducing process produces a tube with mechanical properties beyond those obtainable by the usual cold drawing process. Moreover, no tensional stresses are developed in the tube reducing process and the concentricity of rotary pierced tubes is considerably improved. Also, close tolerances in diameter and ovality are obtained even in very small wall/diameter ratios. Extremely fine surface finishes, inside and outside, are standard conditions. The tube reducing process now in use was discovered accidentally some twenty-five years ago when an experimental machine for tapering golf shafts accidentally jammed and produced a straight tube of greatly reduced cross-sectional area. The process, as it was finally perfected, produces a cold finish tubing in a way entirely different from the usual methods of cold drawing or swaging. It can, in fact, be described as cold pilgering, but with the tube remaining stationary and the roll housing moving back and forth in reciprocating motion. The mandrel against which the tube wall is rolled, is tapered towards the finishing end of the pass to permit the tube to feed free of the mandrel as it elongates in the outgoing direction. The roll pass is elliptic rather than circular and, in order to prevent ovality of the tube and the formation of fins, the tube is rotated about 60 deg. after each pass. The largest tube reducing machine of this type so far in operation takes ingoing tubes as large as $6\frac{1}{2}$ in. diameter but a unit to take 18 in. tubes is under construction.

● *SPOT-WELDING BERYLLIUM-COPPER PARTS*

Beryllium-copper alloy can now be successfully resistance-welded in a spot-welding machine. This achievement does away with an obstacle which restricted the use of these alloys, since until recently the metal could not be welded without loss of its major characteristics of conductivity and resiliency. The solution was found when in a given case the use of beryllium-copper was required for electrical reasons, while at the same time recourse to soldering or riveting was barred by design considerations. The apparatus used for the spot-welding of the alloy is a spot welder of the condenser-discharge type. Actually the machine is capable of storing electric charges at 3000 V in a bank of condensers and then discharging the condensers through a transformer, the secondary winding of which is connected with the electrodes. At maximum output the current at the electrode tips is 23,000 A at 10 V. In spite of this high amperage no excessive heat is generated and a weld is finished in less than a fiftieth of a second. When welding beryllium-copper strips of 0·004 in. and 0·008 in. respectively to a base material consisting of 65-35 bronze, the welding machine is set to store at 2600 V in a condenser of 240 microfarads, resulting in a welding current of 18,000 A at 12 V. Because of the toxic effect of the alloy the welding fumes must be removed by a special exhaust system.

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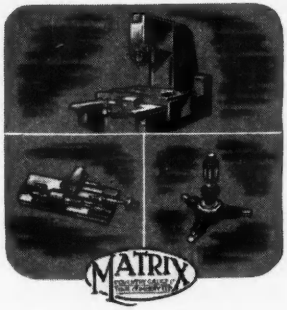

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THE ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

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The Development of the Free-Piston Engine

THE freedom of the free piston, like many other freedoms, has aroused considerable controversy. In reality, it comes nearest to Henry Ford's definition of freedom when, in the old "T" model days, he said that his customers were free to choose the colour of their car provided it was black. The free piston is actually under precise control, ensured by the gentle guidance of the control gear and not the unnatural constraint of the crank mechanism.

Free-piston engines have been under development for some time and hence this review is addressed to the disappointed as well as to the uninitiated and will not easily satisfy both. Nevertheless, each may find profit in an approach which seeks to distinguish the significant from the accidental. In our age, the permanent revolution in technology is not challenged in principle. However, when everything is potentially changeable, it is all the more desirable, as it is more difficult, to know which changes are worthwhile.

The free-piston engine is a prime mover of the internal combustion type. Its essential feature is the direct action of a power piston upon a compressor piston. It is therefore fundamentally a pressure transformer of the oscillating displacement type. One essential feature of the free-piston engine is the achievement of perfect mass balance with two piston assemblies moving in precise opposition. The main consequence of the freedom from positive kinematic constraint is the fact that the stroke of the pistons is controllable in operation. The inevitable corollary of this freedom is that the dynamics of piston motion are determined by the natural properties of the oscillating system, consisting of the piston mass and the elasticity of the air trapped by the piston assembly. These are the only *fundamental* features of the free piston principle.

In practice, several additional features are nearly universal. Most free-piston engines have a straight reciprocating piston motion. It is not possible to maintain the desired conditions in the power cylinder throughout the range of control without introducing an additional air cushion cylinder, in which a quantity of air is trapped, to be changed only during a control action. The precise symmetry in the movement of the opposing piston assemblies must be assured by a positive kinematic linkage known as the synchronising gear. The absence of any motion at the dead points imposes the choice of fuel delivery from a storage system. Most free-piston engines have a power cylinder operating on the two-stroke diesel principle. Existing free-piston engines differ amongst themselves mainly in the lay-out of the power, compressor, and cushion cylinders; in the embodiment of the synchronising linkage; and in the design of the fuel injection equipment and of the control gear.

If the air delivered by the compressor cylinders is the desired output of the machine, the free-piston engine is a pressure air generator or "free-piston compressor." If the pressure energy of the air is to be converted into mechanical work in some form of rotary machine driving a shaft, the free-piston engine becomes a power gas generator or "gas generator" for short.

In the power gas generator it is desirable to utilise in the compressed gas all the available exhaust heat, and this can be done without adding further heat by mixing the compressor output with the exhaust from the power cylinder. For this reason, the power cylinder operates at a high pressure level, the compressor therefore acting like the scavenge blower of a two-stroke diesel engine. This condition encourages and permits much higher combustion rates in the power cylinder, often about 5 times those in a normal diesel cylinder.

The period of gestation in free-piston development has been so long that its paternity is difficult to establish. It is easy to see, however, how the free-piston idea arose. The crank mechanism has been an unwelcome associate of heat engines ever since their inception. James Watt tried to circumvent it because, in his day, it was protected by patents. His solution had few technical merits and was soon discarded. In the second half of the 19th century, probably the most numerous class of patent applications was that concerned with an endless variety of rotary displacement engines. In spite of this profusion of inventive genius, none of the rotary displacement mechanisms has achieved any enduring success in the heat engine field. The nearest approach to the free-piston principle was the direct-acting boiler feed pump and the Humphrey pump. In the latter, a column of water is the free "piston" of a gas engine cylinder, driven outwards by the expanding gas and delivering part of the substance against the pressure head to be overcome, and renewing its contents from the reservoir during the return stroke.

This retrospect shows that the desire to retain what might be called the fluctuating principle in internal combustion prime movers, and simultaneously to eliminate the crank mechanism, has been felt for a long time.

The earliest claim to inventorship in recent literature has been staked by Professor Junkers, who is said to have been at work on free-piston compressors since 1911. British patents date from the early 20's, but there is little doubt that the most determined driving force of this development from 1922 onwards has been M. Pescara, in whose name the principal lineage of this development is continued in France and Britain.

Present embodiments of the free-piston engine start with free-piston compressors. Before World War II, Junkers had a four-stage compressor delivering 70 cfm of air at a pressure of about 3000 psia. This unit was standard equipment in the German Navy for torpedo launching. Several other Junkers models were available for industrial purposes even before then.

The SIGMA organisation in France also had a free-piston compressor for industrial pressures and about 200 cfm output. A similar unit, now offered in Great Britain by Messrs. Alan Muntz and Co., delivers up to 105 cfm at a pressure of 100 psia. The French unit has been built by the hundred since the war.

Owing to the higher combustion rates in the power cylinder of the gas generator, development had to await the solution of problems in cylinder-wall lubrication and piston-ring wear. The credit for the first successful unit must go to Robert Huber working with

Pescara for the French SIGMA Organisation. The SIGMA GS-34 unit delivers gas at 50 psig and 507° C, and has an output of 1310 gas hp at full load, though 1480 hp has been reached on test. The layout of this unit (THE ENGINEERS' DIGEST, Sept. 1950, page 320) is of the single-acting, inward compressing type of compressor piston. Its outward face acts as a cushion piston. Its diesel cylinder has a bore of 13·4 in. and its compressor cylinder a bore of 35·4 in. Its piston stroke is 17·5 in. and its cycle frequency is 600 cpm. Its tested gas efficiency is 44 per cent, corresponding to a specific fuel consumption of 0·32 lb/gas hp-hr.

This unit is unquestionably the most thoroughly developed of all existing power gas generators and is at present in production for the marine propulsion, power station and rail traction applications mentioned below. It was deliberately designed with conservative stresses and temperatures, for low cost of construction and maintenance.

Since the last war, two U.S. firms, with the assistance of the U.S. Navy Bureau of Ships, have developed power gas generators. The Cooper-Bessemer Model R is designed as a heavy-duty low-cost unit. Nevertheless, its gas delivery pressure is increased to 72·5 psig, whilst its temperature of 538° C is raised slightly above the GS-34 value. Its cylinder layout is of the type in which the compressor piston is single-acting, outward compressing, and a separate cushion cylinder is arranged outside the compressor cylinders. The bore of the power cylinder is about 14 in. and that of the compressor cylinder 37 in. At full load, the stroke is about 18·5 in. and the frequency 555 cpm. The rating is 1750 gas hp and the gas efficiency is 45 per cent. Like the SIGMA gas generator, its overall weight is such that, together with the gas turbine plant, the complete installation weighs about 30 lb/hp.

The other American unit which has already undergone tests is the Baldwin-Lima-Hamilton Model B. It has the same cylinder layout as the Cooper-Bessemer unit, but is designed for a further increase in gas delivery pressure to 90 psig and a substantially higher gas delivery temperature of 705° C. This machine has a power cylinder bore of 8·25 in., a compressor cylinder bore of 23 in., and an outward cushion cylinder bore of 8·25 in. Its full-load stroke is 11 in., and its maximum frequency is 1035 cpm. It has an output of 885 gas hp. The higher operating pressure and temperature lead to lower weight per horsepower, or roughly half that of the two former models. Both the Cooper-Bessemer and the Baldwin-Lima units have done some hundreds of hours of test running, but are not yet in production.

A much smaller unit (US-75), is under development in Great Britain by Alan Muntz and Co. Ltd. This unit has the SIGMA layout and roughly the same operating conditions, but an output of about 450 gas hp. Its power cylinder bore is 7·5 in. and its compressor cylinder bore is 20·75 in. This machine works at full load with a stroke of 10 in. and a frequency of about 1100 cpm.

A totally different line of development has been pursued in France by M. Mercier, supported by the French Electricity Authority. His free-piston unit is a rotary displacement type, which must at present be regarded as a daring outsider and is outside the scope of this article.

In view of these achievements, the failures of some organisations which have started similar developments must be regarded as curiosities. Indeed, in one discussion a former devotee of free pistons blamed the elusiveness of M. Pescara.

It was felt that the existence of practical units of the free-piston type needed some emphasis before discussing the advantages of the free-piston engine. It is in the practical and mechanical planes that those advantages which distinguish the free-piston from *all* other engines are found. These special qualities are often indispensable, if "compound" heat engine plants are to be practical. The thermodynamic advantages of the projects mentioned below are due *mainly* to the principle of compounding, though the free piston contributes its share.

The main drawbacks of the crank mechanism are unrelated to efficiency. This mechanism communicates all pressure fluctuations to the engine structure and its shaft. Hence, only the free-piston reciprocating pressure transformer is entirely free from vibrations. With a crank mechanism, a side thrust on the piston is inevitable. The manufacture and repair of elements of the crank mechanism, mainly the crankshaft itself, especially when the crankshaft is large and many cylinders are involved, requires equipment of quite a different category from that needed for the manufacture of cylinders and pistons. Cases are known when a stand-by power station with diesel engines had to be rejected because, in spite of the manufacturers' guarantee to halve the specific fuel consumption of a steam plant, no repair facilities were available for thousands of miles—a common occurrence in remote localities. Visitors to the French manufacturing plant, where free-piston gas generators are made, are amazed at the absence of large machinery.

Apart from absence of vibration, low cost and simple maintenance, the elimination of the crank mechanism permits the choice of higher mean piston speeds. Thus, the swept volume is greater for an equivalent bore and stroke of the power cylinder. This enables operation with large amounts of excess air, thus reducing the gas delivery temperature and permitting the burning of fuels other than diesel oil. Petrol and even residual oils have been successfully used in free-piston engines. Another impressive advantage is almost instantaneous starting with the help of a single charge of compressed air. Within a matter of a few seconds, the generator reaches its full speed.

Considering the combination of a free-piston gas generator with a gas turbine for driving a shaft, no other system embraces all these advantages and yet has an efficiency comparable with that of a diesel engine, retains that efficiency at part load, and does not increase the total installation weight above that of a simple gas turbine plant. All these advantages are equal to the fundamental feature of free piston motion. The diesel engine efficiency is obtained in this compound installation, only because the additional losses, inseparable from compounding, are counter-balanced by the absence of crankshaft friction and by the fact that peak pressures are not limited by crank mechanism design. The excellent part-load behaviour is due to the variable stroke, and, finally, the low weight, though mainly the result of the compounding principle, is due in part to the absence of the crank mechanism.

In listing the advantages, the greatest care has been taken to segregate those inherent in the free piston conception from those which it shares with other forms of compound plants. When the drawbacks of the free-piston engine are considered, it is even more important to distinguish between inherent and accidental faults. Critical sifting of the arguments which have surrounded free-piston engine development leaves only a few fundamental limitations. Some current difficulties and teething troubles will be mentioned below in discussing immediate development work.

Two properties of the free-piston engine may be fundamental limitations. One is the dependence of its piston frequency on other design parameters, in practice mainly on piston mass. Without departing from present design practice, it is impossible to have much influence on the frequency, though it would seem to be desirable in the interest of higher power output. Light alloy pistons and possibly even titanium may alter the situation. The second property is the loss of efficiency, beginning at about 0·25 full load.

Finally, the free-piston engine shares with all piston engines the limitations of being confined to gas and

liquid fuels. Solid fuels can be used only after gasification.

The real value of any machine can be established only by examining its application in concrete cases. The first application of the free piston engine is the free-piston compressor. Here, as mentioned above, the free-piston engine is already a fully developed unit as trouble-free as any mechanical piece of equipment. The free-piston compressor claims lighter weight for the same duty (a reduction of about 25 per cent); a lower fuel consumption (1·9 lb fuel per 1000 cu ft of air at about 100 psig against 2·2 for the crankshaft version); and simpler maintenance. The list is impressive and of the greatest importance for portable compressor units. However, marginal advantages in a tool used under many varying conditions cannot be decided by analysis or discussion. It seems that, in spite of the drawback of high initial cost facing every innovation, portable free-piston compressor units are gaining ground wherever they have been put on the market.

The first use to which free-piston power gas generators have been put is to drive a stationary alternator linked to the French grid for the first time in June 1951. The advantages of the free-piston gas turbine combination for power stations are those of other compound plants. To these must be added the lower cost of the gas generator, especially because of cheap foundations and potentially lower maintenance cost and ability to use cheaper grades of liquid fuel. The Cooper-Bessemer Corporation, in a recent paper, estimates the cost of a generating station of 10,000 kW to be $1,820,000, or about $182/kW. The cost for a steam power plant of the same efficiency is considerably higher.

Like any good compound scheme, the free-piston/gas turbine installation has excellent part-load characteristics down to about ⅓ full load and instantaneous starting, and is therefore suitable for peak load and stand-by stations, whilst the steam plant scores for base load stations only on account of the cheaper fuel it uses. An advantage peculiar to the free-piston gas generator is a station layout of the greatest flexibility, since the gas generators can be located anywhere, and the connection to the turbine is through gas conduits only. Compound projects of comparable efficiency require some mechanical connection between a two-stroke diesel and a gas turbine.

This flexibility is of particular importance for portable generating stations. The free-piston/gas turbine combination is the only one which allows an easy breakdown into separate light units. In the United States the use of free-piston units for pipe-line pumping stations seems especially attractive. The Cooper-Bessemer Corporation estimates the cost of a 10,000 hp pumping station to be $148/bhp.

The use of free-piston/gas turbine units for ship propulsion has been reported in these pages (ENGINEERS' DIGEST, Sept. 1950, page 319). The particular advantages applicable to ships are greater emphasis on lack of vibration, the fact that the compressors are not stopped for reversing, and the availability of overload capacity comparable only with that of a steam-turbine plant. The installation of additional standard units for special conditions is possible only with the free-piston plant because there is no danger of dynamic interaction. At present, a programme of fitting eleven ships with free-piston plants is being completed in France. According to the Cooper-Bessemer Corporation, the installed cost of their heavy-duty type free-piston/gas turbine machinery in 9,000 hp units is $158/hp, compared with $180/hp for steam turbine drive and a still higher figure for diesel drive.

The first trial runs of a free-piston/gas turbine locomotive were carried out last year with great success. The particular advantages of the free-piston plant in this application over all other existing types of locomotives are numerous. Briefly, no other locomotive approaches the successful compromise between first cost, maintenance, and fuel cost, as demonstrated by the French experiment. No other projected locomotive has much chance of doing so, with the exception of the coal burning gas-turbine locomotive, the future of which is still uncertain. Compound schemes, beside the free-piston type, could conceivably have a similar advantage over existing types, but would suffer some lack of flexibility in layout and have less chance of burning low-grade liquid fuel. In fact, the locomotive application appears most attractive even in the United States, where the diesel-electric locomotive is already a considerable advance on existing steam-driven types. It has recently been announced that General Motors has signed complementary agreements with Alan Muntz & Co. Ltd., SEP, and SIGMA, and it is presumed that interest in locomotive application is uppermost in this development.

The road transport application is, at the time of writing, only a project, but the high torque characteristics and quick starting of the free-piston/gas turbine prime mover make it attractive for heavy road vehicles. It is said that the free-piston gas generator can be scaled down to 120 hp and that an installation of this type will require only two forward gears and one reverse.

In this review of the general potentialities of the free-piston engine, many of its details have been purposely omitted. In every mechanical development there is a number of secondary problems, the solution of which forms vital links in the chain of achievement. These problems require resourcefulness and patience but, nevertheless, remain secondary. Every other development likely to achieve the same results would throw up another set of secondary problems, usually unforeseen. In the progress of the free-piston generator, many of these problems have been fully solved. These include cylinder cooling and piston-ring problems, fuel injection, control problems and the opposite phasing of two generator units to feed a single turbine. Other details need further improvements. A major improvement is now under way in the design of high-speed automatic valves. The problem of noise needs further attention, especially suction noise.

Very great advances in power output will be possible by inter-cooling the cylinder charge, by higher operating delivery pressures, involving higher delivery temperatures, and by increased cycle frequencies. Such advances will create new problems of cooling and stability. Great savings in weight are conceivable with more refined and more daring design. In short, the free-piston generator is still in its infancy.

The case for the free-piston gas generator rests on two foundations. The first is the general thermodynamic supremacy of the principle of compounding of what we may call a displacement transformer with a rotating flow machine. This foundation is based on principles of thermodynamics so well established that it is unlikely to be overthrown by any invention in the foreseeable future. It would be wrong to say that every compound scheme is better than any simple scheme, but it cannot be denied that for every simple scheme there is some compound scheme by which the simple scheme is improved.

The second is that the free-piston generator belongs to a particular class of compound prime mover. It has been called the balanced compound engine, in which the displacement component supplies its own supercharge exactly. In this class of compound heat engine the free-piston/gas turbine combination is thermodynamically unsurpassed and has a number of fundamental advantages which will secure for it a very large share in future progress.

Let us hope that this development will be pursued vigorously and with adequate backing, to make sure that we do not, like the proverbial tramp, judge the advantages of a feather-bed after spending a night sleeping on a feather.

Noise Investigation of Rolling Bearings

By G. LOHMANN. (From *Konstruktion*, Vol. 5, Nos. 2 and 3, 1953, pp. 38-46 and pp. 85-90, 23 illustrations.)

This article describes test procedures and equipment employed in investigating the running noise of rolling bearings. Particular endeavours have been made to correlate frequencies with the sources of noise generation. The methods used proceed from the simplest practical workshop tests to investigations with modern electronic and photographic equipment. Procedures for noise reduction are given both for mechanical applications and electric motors.

AN increasingly frequent requirement that machine manufacturers nowadays are asked to fulfil is that their machines should not only give efficient performance but also be capable of operating with a minimum amount of vibration and noise. This requirement is justified, firstly, because it is necessary to reduce noise annoyance and, secondly, because even hardly perceptible mechanical vibrations such as those associated with noise may have an unfavourable effect on the accuracy of a machine operation.

Noise is the result of air vibration acting on the human ear. To reduce this air-borne noise, the machine designer has to investigate the primary sources of the noise, i.e., the mechanical components vibrating as a result of impact, rubbing or accelerating forces. In spite of extreme care taken to obtain smooth running of the moving parts, a certain amount of mechanical vibration will remain inherent in the operation of the machine, and it will be transmitted to the bearings. Thus, one may always expect structure-borne noises and vibrations to occur in bearings.

As far as possible, structure-borne noise should be damped out or isolated before it is radiated to the surrounding air. It is important to avoid the occurrence of noises in the range of frequencies between 1000 and 5000 cps to which, from the point of view of loudness, the ear is most sensitive. Moreover, it should be noted that noise radiation increases with frequency.

The requirement for rolling bearings with minimum noise generation is particularly important in the electric motor industry. In electric motors, the vibratory forces to be dealt with may be of magnetic origin. Generally, the running noise generated in bearings is only a small proportion of the total noise. If this is not the case, there must be considerable vibratory forces acting on the bearings, and these must be reduced by suitable means.

It should be pointed out that it is not easy to alter the running noise of a rolling bearing. In order to have easily interchangeable components, it is not possible to alter the external dimensions substantially or, owing to manufacturing reasons, the internal structure. Thus, to produce a change in the vibratory behaviour of a bearing, we can alter only the surface qualities, tolerances and bearing clearances. Although nowadays, in many cases, surface quality and manufacturing accuracy have reached the highest level permitted by cost considerations, and the values for tolerances and bearing clearances are the most favourable that can be obtained from practical experience, the fact remains that mass-produced bearings, manufactured under apparently identical conditions, do not all fulfil the requirements for quiet running, and there is no clear reason why this is so.

Both manufacturers and users of rolling bearings have endeavoured to develop reliable and as simple as possible test procedures for determining the running noise of rolling bearings. In view of the complex nature of this running noise, it is, however, understandable that the subjective assessment by ear is uncertain. It is, therefore, desirable to have an objective method of testing, which will indicate true noise values and enable a study to be made of the causes of running noise in rolling bearings. In the following, details are given of investigations undertaken in this connection in the bearing test laboratory of The Kugelfischer Georg Schäfer Co., of Schweinfurt, Germany.

TEST RIGS AND TEST METHODS

It should be noted that the influence of the test rig on the bearing to be tested cannot be entirely avoided, but it should be as small and as constant as possible.

The two possible types of test rig are basically as follows :—The first arrangement comprises a shaft supported in two plain bearings. At one end, the shaft is driven by an electric motor through a flexible coupling or a belt-and-pulley system. The other end has an overhanging portion on which the test bearing is fitted. Rotation of the outer ring is prevented either by means of a pad subjected to pressure from a spiral spring, or by a strap placed round the outer ring and connected to a suitable weight.

The second arrangement comprises an electrically driven shaft, which is supported by a vee-type plain bearing near the driven end and has the test bearing supporting it at the free end. The intermediate shaft portion between the two bearings should be fairly long to reduce the effect of running irregularities of the plain bearing on the rolling bearing to be tested.

In both arrangements, thrust collars are provided to avoid axial motion. The fabric belt or coupling should be soft enough to damp any vibrations due to the electric motor. The entire rig should be accurately designed. The basic difference between the two arrangements is that, in the second test rig, the rolling bearing is tested under conditions more closely resembling those actually occurring with a fitted bearing, the amount of motion of the outer ring being limited only by the diameter of the bearing supporting bracket. On the other hand, in the first arrangement, the running inaccuracy of the test bearing has no effect on the shaft, since the latter is supported in two independent bearings. Owing to considerations of accessibility, the second test arrangement is, unfortunately, rarely used. For simplicity of removal, etc., the test bearing is frequently fitted only with a push fit or a tapered fit, although the fitting should be as solid as possible without causing any errors due to distortion.

The measuring equipment should be suitable for the indication and evaluation of very low amplitude vibrations in the acoustic range of frequencies. Basically, it comprises a pick-up, an amplifier, and a monitoring or pointer-type measuring apparatus. The pick-

Fig. 1. Test rig with overhung bearing.
a = Shaft ; *b* = auxiliary bearings ; *c* = test bearing ; *d* = motor with flexible coupling : *e* = alternative belt drive ; *f* = rotation check using strap and weight ; *g* = alternative check using pressure pad.

up is designed as an electro-acoustical " transducer," i.e., a unit which converts air-borne or structure-borne vibrations into electrical signals.

If air-borne noise is to be measured, the tests should take place in an acoustically insulated chamber. The driving motor should be outside this chamber and the drive shaft should be connected with the test rig through an acoustically insulated gland in the wall. This is necessary if low-noise bearings are to be tested, since in such a case the extraneous disturbances causing a " background noise " should be less than 30 phons.* To obtain a sufficiently high input, the microphone-type pick-up is placed as close as possible to the test bearing, but not so close as to be affected by air eddies or disturbances in the sonic field. It should be noted that the shape of the test chamber affects the sonic field and, hence, the noise measurements. Owing to these difficulties, air-borne noise measurements are not very frequently carried out in bearing tests.

For measurements of structure-borne noise, the transducer used is a vibration pick-up of the spring-and-mass type, which is in contact with the vibrating component. The frequency range which is mainly of interest is in this case between 500 and 5000 cps. A fully suitable pick-up for structure-borne noise indication is not yet available. Moreover, although the background disturbance in the form of air-borne noise is eliminated, there is a background vibration due to the components of the test rig and the disturbances transmitted by the foundations, but fortunately it is at a fairly low level.

VIBRATION MEASURING EQUIPMENT

A typical pick-up employed for indicating structure-borne vibrations is the " electrodynamic vibrometer." Clamped round the inner periphery of a rigid cylindrical casing is a diaphragm (used as a spring element), to which a moving-coil system and a feeler rod are connected (these form the " mass " of the system). Displacements of the rod also move the induction coils over the poles of permanent magnets arranged in the casing. The vibration thus produces induced voltages in the main coil, and eddy-current damping is obtained with the auxiliary coil. The signal voltages are proportional to the vibration velocity dy/dt.

The transducer output may or may not depend on the frequency of the vibration being indicated. For instance, the above electrodynamic vibrometer is frequency-dependent; this is also the case with other electromagnetic and magnetostrictive units. No frequency-dependent effects are obtained with capacitive, piezo-electric, carbon-pile, and other types of pick-ups. However, there is another manner in which frequencies have to be considered. A displacement-measuring device must have a natural frequency much lower than that of the vibration which is to be indicated, whereas the reverse applies to acceleration-measuring devices.

Basic requirements for all pick-ups are that they should have a high sensitivity, a linear response over a wide range of sonic frequencies, and that they should not introduce distortions in waveform or phase errors. For indicating rapidly varying phenomena, quick response is also required, that is, the pick-up should follow the peak values at any instant. The known types of pick-up can fulfil only some of these requirements. If the frequency range to be covered does not exceed 500 cps (or at most 1000 cps), the above-mentioned electrodynamic vibrometer is suitable. For measurements of structure-borne noise in the sound-frequency range, piezo-electric pick-ups used as accelerometers give the best results. Acceleration-sensitive instruments, however, are mainly useful to indicate vibratory forces. For sonic pressures transmitted through structural components, velocity is also a decisive factor, so that both velocity and acceleration type pick-ups can be used with advantage.

The calibration of pick-ups by the user would require a considerable outlay of special equipment (vibrating platforms, etc.) in order to simulate adequately the composite vibrations occurring in practice. Comparative performance tests of the various standard types of pick-ups would therefore be a valuable contribution to noise investigations.

For measurements of structure-borne noise, the feeler rod of the pick-up is placed in contact with the outer ring of the test bearing. If the test bearing is on an overhanging portion of the shaft, as shown in Fig. 1, the arrangement presents no difficulty. For a test rig in which the rolling bearing also serves as a supporting bearing, it is necessary to provide a passage to the outer ring through the bearing bracket. The pick-up can either be fixed rigidly on the bracket, in which case it indicates relative motion, or provided with an elastic suspension so as to indicate absolute motion, as illustrated in Fig. 1. In all tests described in the following, this latter arrangement was used and only absolute measurements were taken.

The output signal of vibration pick-ups also requires amplification. It is advisable to check the frequency response of the valve type amplifier. In some cases, departures from linearity are intentional, in order to compensate for certain characteristics of the pick-up or to produce such characteristics. Frequently, the amplifier incorporates integrating and differentiating circuits, so that a velocity-type pick-up may also be used to obtain values of displacement and acceleration. For visual observation of noise variations, the cathode-ray oscillograph is generally used. The galvanometer-type oscillograph is limited by its fairly low frequency and the amount of illumination available. For recording, single photographs can be taken for transients, or films for longer processes.

The analysis of recorded data is not simple. To avoid involved harmonic-analysis evaluations, the usual procedure is to employ a frequency-selective circuit or filter unit. These electronic analysers, connected between the pick-up and the amplifier, are generally of two types: the octave type filter (with fixed settings and a large band width at each frequency setting) and the adjustable RC resonance circuit, which gives a much better selectivity. With filters of this type it is possible to separate peaks at fairly close frequencies. However, this requires a certain amount of time, and if the frequency-spectrum is varying, as is frequently the case with running noise, the analysed result becomes rather uncertain.

Finally, it should be mentioned that improved results may be attainable with amplitude-modulated carrier-frequency equipment. This equipment allows static calibration to be obtained, as well as frequency-independent operation over a wide frequency range. For example, a capacitive system can be employed, with a fixed condenser plate mounted close to a rotating shaft which is under investigation (see Fig. 2). These two elements form a measuring condenser and are included in one branch of a bridge circuit, supplied with carrier frequency by a h.f. oscillator. Coarse and fine compensation are by means of the trimming condenser c and the resistance d. The difference voltages of the bridge, due to a vibratory input, are fed into a valve amplifier and indicated by an oscillograph f.

*Sound pressures are determined in " microbars " or dyne/cm² by means of the relation:

$$p = \rho c u \qquad \text{[microbars]}$$

where ρ = mass-density of the air [gr/cm³], c = velocity of sound [cm/sec] i.e., propagation velocity of sound-waves, and u = phase velocity of the noise [cm/sec]. Related to a reference signal at 1000 cps, the subjective effect or " loudness " L of the noise is then obtained from

$$L = 20 \log (p_1/p_0) \qquad \text{[phons]}$$

where p_0 is the lowest audible sound pressure (" audibility threshold "), used as a reference value.

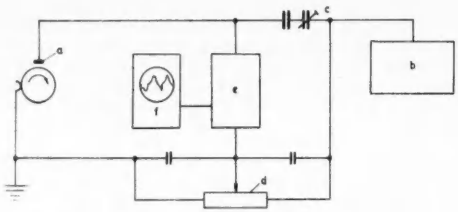


Fig. 2. **Amplitude-modulated carrier-frequency system with capacitive pick-up.**
a = pick-up ; *b* = h.f. oscillator ; *c* = trimmer ; *d* = variable resistance ; *e* = amplifier ; *f* = c.r. oscillograph.

GENERATION OF RUNNING NOISE IN BEARINGS

It is of interest to consider the possible causes of the frequency peaks in the noise spectrum of rolling bearings. Although the manufacturing tolerances set forth in a German Standard Specification are small in relation to bearing size and shape, they should be included in considerations of bearing noise. Generally, it is found that the rings are of suitably circular shape. However, a slight deformation of the cylindrical inner surfaces cannot be entirely avoided, since these surfaces go through two machining operations. Thus, variations in wall thickness are obtained and these, during the rotation of the bearing, produce a radial impact. With, for instance, an inner ring of non-constant wall thickness, a shaft is periodically raised and lowered during its rotation. The plot of displacement amplitudes against time gives a sinusoidal curve, with a frequency of $n_i/60$ [cps], where n_i is the rpm value for the inner ring. This curve represents the running accuracy of the bearing. If the inner ring diverges at several positions from the true circular shape, the sine curve will show periodically similar deviations.

This running-accuracy curve is modulated by the motion of the rolling bodies. Magnifying the process, it may be said that the shaft receives an impact each time a rolling body passes under it. These impacts occur at a frequency Z which is termed the " ball interference " frequency. Z is obtained as the product of the cage rpm, n_{ca}, and the number of balls in the bearing. The cage speed can be determined from

$$n_{ca} = (n_i/2)[1 - (d_w/T)] \quad [\text{rpm}]$$

where n_i = rpm of inner ring, d_w = diameter of rolling bodies, and T = pitch circle diameter of the rolling bodies. In practice, therefore, n_{ca} is not quite equal to half the operating speed. In most cases it is approximately $0{\cdot}4\ n_i$.

A further modulation of the sinusoidal vibration may occur if the balls are of unequal thickness. Diametral deviations of bearing balls are of the order of 0·001 mm. If a ball is not absolutely spherical, its impact on the shaft will depend on its reaching the same angular position after performing a certain number of revolutions in the raceways which meet it at the same relative positions as for a previous impact. These conditions can be expressed mathematically, but for our purpose an illustration given in Fig. 3 is sufficient. It should be noted that irregularities of this kind can hardly be expected to give sharp frequency peaks. Similarly, surface roughnesses mainly result in irregularly varying disturbances.

Various non-periodic vibrations may occur in addition to the periodic vibrations described above. Rolling or sliding friction in the cage or the races, imperfect rolling conditions and slipping in the unloaded sections of the outer ring race are possible causes of irregular vibration. Cage impacts may be due in some cases to excessive quantities of air in the recesses of slowly rotating bearings. Since low-noise bearings are the purpose of this investigation, such factors are of appreciable importance in connection with " cage noise."

Two further factors of importance in regard to running-noise and structure-borne noise are (i) the presence of dirt and grit, and (ii) the running inaccuracy of the test rig. In contrast with practical operating conditions, all our tests were made with absolutely clean bearings. As regards the test rig, no matter how accurate it may be, it inevitably produces additional radial impacts on the test bearing.

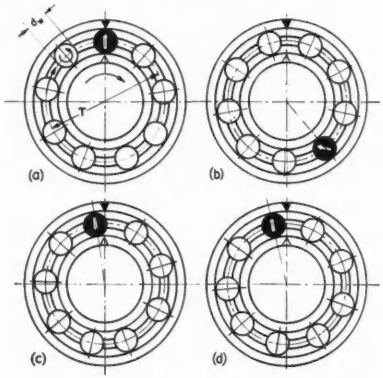


Fig. 3. **Rotation of balls in a 25 mm bore bearing.**
(*a*) Initial position ; (*b*) position after one full revolution of inner ring ; (*c*) position after 5 revolutions ; (*d*) position after 10 revolutions. Outer ring held stationary. $T = 38{\cdot}5$ mm ; $d_w = 8$ mm.

DESCRIPTION OF TESTS

" *Subjective* " *tests.* For comparative purposes, it is always advisable to begin an investigation of bearing noise with subjective tests, which serve to determine the impression made by the noise on the human ear.

The arrangement used is generally similar to the rig shown in Fig. 1. The investigator, while listening to the noise generated, presses gently with one hand on the outer ring of the test bearing in the axial direction. This enables him to assess by feeling the smoothness of the running conditions. It is remarkable how rapidly a trained assistant can pick out the best and the worst bearings from a not too large quantity being tested. Unfortunately, he cannot differentiate further with sufficient reliability between these extremes. To assist his judgment in this respect, the tester listens from time to time to the running noise of bearing specimens which have been found normal or still good enough. In spite of this, the assessment by ear is always subject to variations depending on the mental or physical condition of the tester. The reference noise can be recorded on a magnetic tape, which is then used as an endless band in a reproducing system, with a loudspeaker which can be switched over from the reference noise to the output noise of the test bearing. It is important to use the same electro-acoustical equipment in both cases.

Air-borne noise measurements. The next step is to obtain quantitative indications by objective measurements. For this purpose, the test rig in Fig. 1 was used, with a capacitive microphone at 85 mm in the axial direction away from the pitch circle of the rolling bodies of the bearing under test. Rotation of the overhung bearing was prevented by means of a strap carrying a weight of 5 kg. Three sets of four or five bearings were tested, one set consisting of " normal " bearings, and the other two of "low-noise" ball bearings, all of 30 mm bore and of single-row, deep-groove type.

In view of the background noise of 53 phons, due to the test rig, the first series of tests was made with unlubricated bearings, moist only through being rinsed in

petrol. The noise levels were between 60 and 70 phons at about 1420 rpm. For the second test series, the bearings were lubricated with four drops of spindle oil and run at 1420 rpm. The running noise decreased to 53-56 phons. In a third series of tests, the bearings were run at 3050 rpm, after being lubricated as in the former test. The background noise level increased to 63 phons and the bearing noise was between 64 and 68 phons. Although the low-noise bearings are precision bearings, the measurements of air-borne noise did not indicate much difference in loudness between these and the normal bearings. In fact, these air-borne noise measurements were not found satisfactory for the purpose, either owing to the high level of the background noise, or because the air-borne noise did not provide all the relevant information. In any case, this procedure cannot be used to sort out bearings according to their running noise.

Structure-borne noise measurements. Using the same test rig as previously, a velocity-sensitive pick-up was arranged in contact with the upper part of the outer ring of the test bearing, so as to indicate the vertical vibrations. To identify disturbing impacts due to the test shaft, before each subsequent test the inner ring of the bearing was rotated through 60 deg on the test shaft and then secured by a push fit and screwed up tight by means of a nut. At least two measurements were obtained for each bearing position. The electronic equipment included integrating and differentiating circuits, which enabled vibratory displacements and accelerations to be indicated as well as velocity.

It was found that the measured values varied according to the angular setting of the inner ring on the test shaft. Repeat tests gave approximately the same values only for the displacement amplitudes. Further tests were made with the test bearing used as a shaft support. The outer ring vibrations were indicated by a feeler rod placed in a suitable passage in the bearing bracket. Measurements were taken with a vacuum tube voltmeter or from oscillograph screen readings. Considerable fluctuations were found in the results and it was not possible to obtain reproducible values.

In this connection, it is useful to differentiate between the ordinary arithmetic mean and the mean value taking account of the frequency of occurrence of each reading. Moreover, it is sometimes necessary to filter out the frequency corresponding to the running speed, since it is not important as regards noise generation, although it produces appreciable radial motion of the inner ring. This is borne out by the fact that the greatest amplitudes at the fundamental frequency were obtained with the low-noise bearings. Measurements were taken with bearings of different types, covering a wide range of manufacturing accuracies, tolerances and surface qualities. In all cases, it was found that the results were not reproducible and that there was no clear correlation between accuracy or clearance and the running noise. Therefore, this method also cannot be regarded as suitable for noise investigations.

Frequency-spectrum analyses. In the above tests, only the fundamental frequency corresponding to the running speed was filtered out. As the vibration diagrams obtained were extremely irregular, with the exception of the running-speed frequency peaks, it was endeavoured to obtain further information by analysing the frequency spectrum.

In all these tests, the air-borne noise was measured, using a capacitive type microphone. The first tests were made with a *RC* filter with negative feedback. This unit has the disadvantage of 50-cycle interference at low frequencies. Despite some fluctuations, it was found that the readings gave sufficiently consistent values and reproducible results.

A typical spectrum for a ball bearing of 25 mm bore running at 2650 rpm shows that the frequencies of the background noise are generally below the 1000 cps value. Sharp peaks are visible at about 2500 and 6500 cps which are definite bearing noise frequencies. However, no marked peaks (apart from that of the running speed) were obtained in the low frequency range, although for instance the " ball disturbance " frequency is at 113 cps and seemed to be present in subjective tests.

Further frequency analyses of a 35 mm bore bearing were therefore carried out using the test rig in Fig. 1. These tests were made by the Federal Physical Laboratory,* using a Grützmacher narrow band-width heterodyne type analyser. With this equipment, more time is required to determine the frequency spectrum, but the results are more accurate. Mains interference occurs again at low frequencies. The bearing noise was recorded on magnetic tape, which was then used as an endless band for spectrum analysis. Examples of the diagrams obtained are shown in Fig. 4. Background noise is below 1000 cps and a large number of peaks is present from 2500 to 6000 cps extending to 9000 cps in one case.

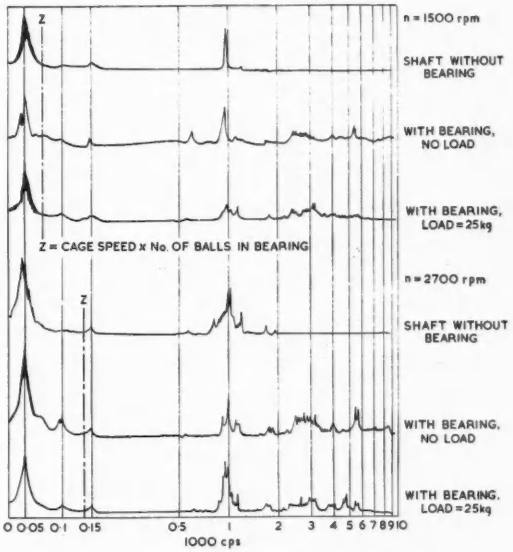

n = 1500 rpm
Z
SHAFT WITHOUT BEARING
WITH BEARING, NO LOAD
WITH BEARING, LOAD = 25kg
Z = CAGE SPEED x No. OF BALLS IN BEARING
n = 2700 rpm
Z
SHAFT WITHOUT BEARING
WITH BEARING, NO LOAD
WITH BEARING, LOAD = 25kg
0 0·05 0·1 0·15 0·5 1 2 3 4 5 6 7 8 9 10
1000 cps

Fig. 4. **Frequency analysis of running noise of a 35 mm bore bearing at 1500 and 2700 rpm.**

In order to determine the frequency of occurrence of these peaks, " visible speech " equipment (originally developed in the United States) was set up for improvised tests at the Federal Physical Laboratory. The recordings are on a strip of light-sensitive paper with time markings. The vertical scale corresponds to the logarithm of the vibration frequency. Each time a frequency occurs there is a corresponding blackening on the paper, and its magnitude gives an approximate indication of the height (i.e., intensity) of the peak. These tests showed some remarkable results: (i) the frequency constituents do not alter appreciably with running speed but fluctuate about fixed frequency values; (ii) their instantaneous amplitudes, however, are extremely irregular; (iii) the frequency constituents of the bearing noise and the background noise decrease in number with increasing running speed; and (iv) the " ball disturbance " frequency is hardly apparent, though it may possibly be more prominent with higher loads.

* Physikalisch-Technische Bundesanstalt (PTBA).

A frequency in the region of 3000 cps was identified in practically all the spectra obtained. This frequency corresponds to the calculated natural frequency of the inner ring. Subsequently, the Federal Physical Laboratory determined experimentally by electromagnetic excitation the natural frequencies of radial and axial vibration of outer rings of various types of bearings with an inner diameter of 35 mm, for one, two and (in some instances) three node vibration conditions. It is of interest to note that these correspond with a large proportion of the frequency peaks of bearing noise determined by spectrum analyses.

PRACTICAL ASSESSMENT OF RUNNING NOISE

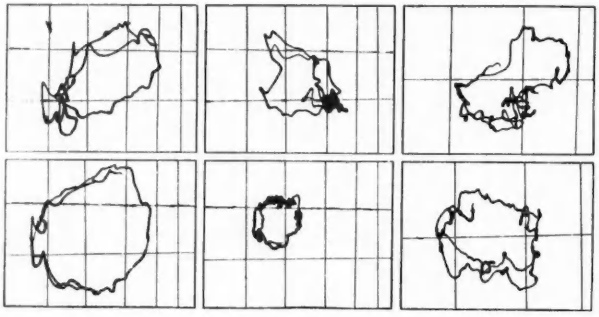
The noise which is described as " running noise of rolling bearings " is due to the irregularities in shape of the rolling elements, which cause vibrations of the shaft in its clearance spaces. These vibrations produce on the outer bearing ring impulses which are transmitted to adjacent machine components (casing, foundation, etc.) as structure-borne noise and then radiated as sound to the surrounding air. It is thus understandable that there is little direct correlation between running speed frequencies and noise frequencies, and that the noise intensity increases with running speed.

To determine outer-ring vibrations, running accuracy diagrams were taken with a capacitive type pick-up arrangement, using the outer ring as one electrode and a stationary plate as the other, in conjunction with carrier-frequency electronic equipment. An amplitude sensitivity of 10^{-4} to 10^{-5} mm was obtained. It was found that the main errors in shape of rolling bearings can be indicated as follows :—

(1) Large sinusoidal vibrations of the order of 0·01 mm are due to eccentricity of the raceway of the inner ring, resulting in radial impact.
(2) Periodic or non-periodic medium and small amplitude vibrations of the order of 0·001 mm, modulating the larger sine variations, are due to small shape irregularities in the machining of the raceways or diameter differences of the rolling elements.
(3) Very small amplitude vibrations, modulating the above, are due to surface-structure irregularities.

The diagrams can be photographed on an oscilloscope screen with a time-base controlled by the shaft speed. By using a two-channel amplifier and a second capacitive element, at 90 degrees to the first, relative to the bearing outer ring, x- and y-axis vibrations can be indicated simultaneously (without a time base) and the impulses amplified and transmitted to the corresponding deflector plates of the oscilloscope. " Polar diagrams " obtained in this manner for 25 mm bore ball-bearings of the single-row, deep groove type, are shown in Fig. 5.

This figure shows that, with a radial load of 3 kg (upper diagrams), the addition of a slight axial load of 2 kg (lower diagrams) considerably improves the running smoothness by preventing excessive "wobbling" of the shaft in its bearing. The diagrams obtained with axial load are much more regular than those without this load. The straight lines in the figures are calibration lines derived from preliminary static calibration tests.



" good " " medium " " poor "

Fig. 5. **Polar diagrams of 25 mm bore bearings at 2150 rpm.**

RUNNING NOISE OF ELECTRIC MOTORS

Electric motors, and particularly small three-phase motors, are increasingly required to operate under quiet-running conditions. In this connection, it is necessary to distinguish between mechanical noises and those of magnetic origin. The fan frequently requires re-designing and the air inlet passages can be provided with shields or baffles to reduce air-borne noise. To isolate bearing noise, it is convenient to run the motor at half its rated voltage and to follow this with running-down tests after cut-off. These remarks indicate that the running noise of bearings is only a part of the total noise.

As regards noise, electric motors can be divided into two main classes, viz. (I) machines with fairly constant noise characteristics and (II) machines with irregular or transient noise characteristics. In motors of class I, the noise can be reduced by removing grit and rinsing the bearing elements, or by replacing unsatisfactory bearings, where necessary. Rough running or humming noises indicate that the radial clearances used are either too large or too small. Moreover, a slight axial pressure on the bearings, by means of coil springs, can produce a beneficial result. In some cases, it may occur that additive tolerances are all on the high, or the low, side. To avoid this, many bearing manufacturers are adopting the practice of sub-dividing bearings into more finely graded classifications. Sometimes, the accuracy of the bearing supports in the motor, or of the outer supports, is found insufficient.

For motors of class II, the above precautions do not provide the full remedy. Strong noise fluctuations and chatter are apt to occur intermittently, as well as howling noises when the motor is running down or is placed in an inclined position. These are nearly always due to vibratory forces acting on the bearings and not originating in the bearings themselves. These vibrations can generate airborne noise directly or cause structure-borne noise of the motor housing.

When such disturbances occur, the electrical parts of the motor should be investigated to ascertain whether the design is suitable for minimum noise requirements, In particular, the number of slots should be carefully chosen, in spite of difficulties, for instance, with change-pole machines. The rotor must be rigid enough and adequately balanced. The latest types of balancing machines give a degree of accuracy which is greater than that obtainable with rolling bearings, taking into account the radial impact effects occurring in the latter. In such cases, it is advisable to balance the rotor with its bearings fitted, in spite of difficulties and somewhat less accuracy in the balancing operation. Structural modifications, such as a stiffening of the motor in the radial and axial directions, may be necessary. If this is not sufficient, vibration damping supports (e.g. with rubber elements) may have to be considered, to avoid transmission of structural noise to the foundation. Vibration damping coatings are also a method of reducing noise radiation to the surrounding air. All these measures represent a corresponding financial outlay and in each case a compromise to suit the users' requirements is desirable.

Noise tests of electric motors are usually carried out by measuring the air-borne

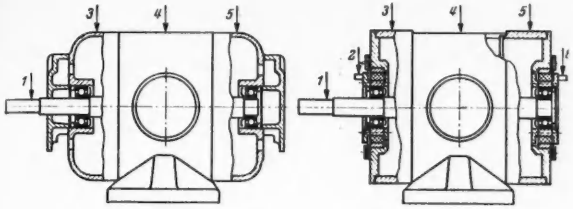


Fig. 6. **Motor before modification for noise reduction (a), and after modification (b).**

noise, with corresponding frequency-spectrum analyses for each structural modification. In the low-frequency range, the spectrum peaks are generally around 50 and 100 cps, and are due to the reversals of magnetisation; sometimes the "bearing disturbance" frequency can be identified. Often there is a frequency equal to the product of the number of slots and the motor speed in rps, which also has harmonics in the medium range of acoustic frequencies. Between 2000 and 8000 cps, the frequencies due to the bearings can be determined. Although their amplitudes decrease with higher frequencies, they are noticeable as disturbances, because they occur in the frequency range of maximum aural sensitivity.

As an example, consider the case of a three-phase motor of 1·1 kW rated power at 2860 rpm, requiring modification for quiet-running conditions. The motor was tested both under normal running conditions and with an external drive. By suitable redesigning, and by providing vibration damping devices for the bearings, the loudness level was reduced from 66-67 phons (against a background level of 30-31 phons) to 54-55 phons (background 33 phons). Figs. 6a and 6b show this motor before and after modification for noise reduction. The numbers in the figures indicate the respective noise measuring positions.

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U.S.A.

The Ideal Automatic Car Transmission

By B. EDSALL.

(From *Automotive Engineer*, Vol. 108, No. 9, May 1, 1953, pp. 66-67, 106, 108, 112, 114, 3 illustrations.)

The major factors in evaluating the requirements of an ideal automatic transmission are discussed. In the author's opinion, such a transmission will necessitate considerable development of, or even deviation from, the fundamental designs in use at present.

WHEN evaluating an ideal automatic car transmission, four major factors must be considered. In listing the four major factors no particular emphasis is intended by virtue of their order of presentation. All are of primary importance to automatic transmission evaluation, although it will be observed that manufacturers themselves have exhibited preference in their product design. These factors are:

(1) Completely automatic performance, characterised by the absence of a clutch pedal and by the lack of requirement to return to zero throttle to obtain an upshift. This feature requires the following:
   a. The ability to start, drive, brake or reverse with no driver action other than manual selection of the transmission control.
   b. The ability in drive range to power-shift through any ratio change and automatically provide the driver with the correct ratio for maximum performance.
(2) Complete smoothness of operation, even when a ratio change is made.
(3) Economy of fuel consumption.
(4) Competitive cost of manufacture.

The ideal automatic car transmission would provide the driver with all of these features without compromise.

Fundamental designs at present in use force an immediate compromise of these four major features, whereas, if it were possible to combine the best features of all existing transmissions into one, the result would be the ideal automatic transmission ("Idealomatic.").

AUTOMATIC PERFORMANCE CHARACTERISTICS

One of the primary functions of such a transmission is to provide maximum full throttle performance automatically. However, even the ideal transmission is a compromise in this case, since consideration must be taken of the engine and its noise level characteristics.

Performance characteristics of an average American passenger car engine, obtained by averaging a number of 1951 passenger car engines, show that the horsepower peak occurs at 3700 engine rpm and the torque peak occurs at 1850 rpm. Also, the maximum hp available varies only four per cent in the 3200-4050 rpm range.

Maximum full throttle performance could be obtained from this engine by providing an infinitely variable transmission with 100 per cent efficiency, that would allow the engine to operate constantly at 3700 rpm, its horsepower peak. However, owing to the objectionably high level of engine noise that would prevail at low car speeds, this transmission, even if it were attainable, would not suit passenger car requirements.

Figure 1 shows this performance characteristic as the upper curve. In order to obtain this curve, an average car weight of 3700 lb was assumed and an axle ratio of 3·0 with a 28 in. tyre diameter. The last two items represent present limitations on the overall power-train economy ratio.

The engine speed curve that could give a pleasing characteristic, combined with performance, is shown in Fig. 2. This shows the engine speed at 1900 rpm, just slightly above the engine's torque peak, at zero car

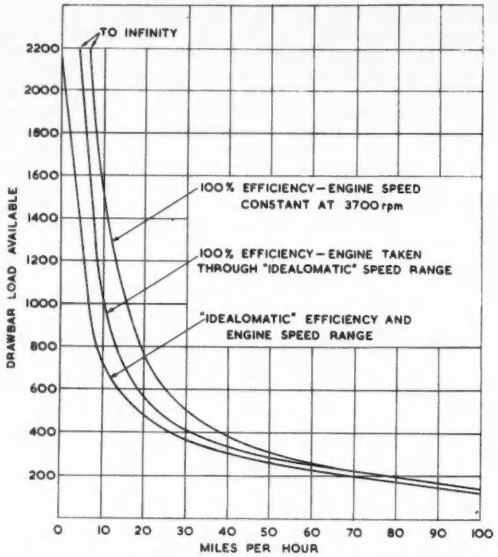

Fig. 1. **Drawbar load with " Idealomatic " transmission, compared with maximum available drawbar.**

speed. As car speed increases, the engine speed rises proportionately, passing through the horsepower peak at 86 mph. It will be noted that within four per cent of maximum horsepower is available for torque conversion from 62 to 100 mph.

Up to this time only 100 per cent transmission efficiency has been considered, but if the " Idealomatic " is to possess the best characteristics of present automatic transmissions, an efficiency curve must be assumed.

The assumed efficiency is shown in Fig. 2, and is the average efficiency of the most efficient fluid-coupling type transmission existing at present.

The maximum torque conversion ratio required at zero car speed would be that which would slip the wheels on a 60 per cent grade. With the axle and wheel ratio previously specified, a transmission ratio of 4 to 1 would more than meet this requirement.

The torque conversion ratio of the " Idealomatic " is, therefore, shown in Fig. 2. Starting with a 4·0 ratio at stall, the conversion is traced throughout the speed range.

The torque ratio beyond stall is the maximum possible with the efficiency and engine speed limitations that have been set.

At this point, it is interesting to note that we have set up requirements that no fundamental design of today can meet. The efficiency curve taken as a target is for a fluid-coupling type transmission, and converter type transmissions fall well short of this mark.

The engine curve that has been selected as optimum can be obtained only with an infinitely variable torque ratio type of transmission such as a fluid torque converter. Gear type transmissions must rise well above the optimum engine speed, so that their average performance in each gear is acceptable.

The stall ratio of 4·0 selected is not obtainable by any of today's torque converters. All must have a geared ratio in addition, which calls for a friction clutch shift and change of engine speed.

Reverting to Fig. 1, the performance of the " Idealomatic " with the average passenger car is shown.

As already stated, the drawbar load, or maximum force at the rear wheels, of the 100 per cent efficiency transmission operating at 3700 engine rpm, is shown as the upper curve.

The middle curve on the sheet shows the drawbar force available owing to engine speed reduction below the horsepower peak. The difference between these two curves may be called the horsepower sacrificed for comfort.

The lower curve represents the " Idealomatic " performance, with both the losses for reduced engine speed and efficiency included.

Reasonable ability in the 60-90 mph range has suffered very little as a result of the " Idealomatic's " engine speed and efficiency. This is a direct result of the small variation in horsepower input by the engine in the 3200-4050 rpm range and the high speed efficiency.

With the performance curve shown, the " Idealomatic " would not provide overall acceleration ability much greater than that of some of the geared types of transmissions today, but it would provide it smoothly and pleasingly.

The maximum full throttle performance of the " Idealomatic " must be available at full accelerator position and at any car speed if the transmission is to be 100 per cent automatic. The ability of the transmission to provide maximum tractive effort and the ability to provide this tractive effort automatically are two separate and distinct functions in present transmissions.

In many cases, particularly where a geared ratio, or a selection of geared ratios, is provided behind a converter, a manual shift must be made to obtain maximum tractive effort at the lower car speeds.

Even some of the present most fully automatic geared ratio transmissions suffer a deficiency in this respect. Although maximum tractive effort is available when the car is accelerated from a standing start, it is not available after the transmission has once upshifted and car speed has been maintained. At some car speeds, automatic downshifts are made into 3rd gear ratio when 2nd gear would provide better performance, or into 2nd gear when first gear performance is more suitable.

The performance limitations of present transmissions may be summarised as follows :

(1) Present converters supply an infinitely variable ratio which covers only a part of the performance range. As a result, one or two mechanically-geared ratios are required in addition to the converter.

(2) No transmission of any type provides 100 per cent automatic performance in the full throttle range, when called upon to supply maximum tractive effort.

(3) In most cases, fully automatic performance could be available, but, to make it more available in present designs would compromise other equally important factors.

Fuel Economy and Part Throttle Performance Characteristics

The " Idealomatic " would cause the engine to operate in its most economical speed range under normal road load and light throttle acceleration conditions. This implies overdrive due to the friction horsepower characteristics of the engine and present limitations on lower axle ratios.

Converters are relatively efficient fluid couplings when driven at light throttles and constant speed. With light and medium throttle acceleration requirements, such as are encountered in normal city driving, their inherent nature is to increase engine speed and decrease their own efficiency, a double loss of economy.

A range of torque multiplication must be provided between the part throttle economy drive and the full throttle performance if the " Idealomatic " is to provide satisfactory overall performance. There is a definite requirement for an infinitely variable torque multiplication, for passing and grade climbing at part throttle conditions. This requirement is well met by present converters and only partially met by an automatic gear range selection.

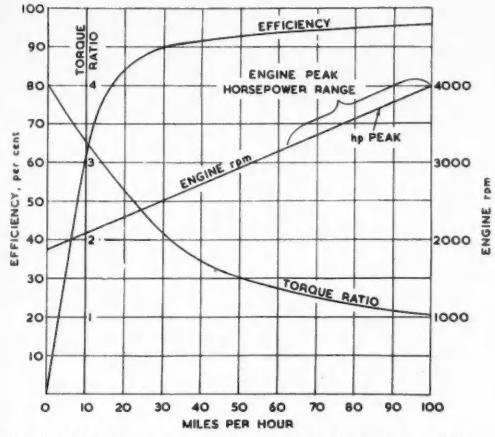

Fig. 2. **" Idealomatic " performance, full throttle. Efficiency, torque ratio, and input speed with average passenger car engine.**

SMOOTHNESS AND RESPONSE

For absolute comfort, the " Idealomatic " must be completely smooth in its transition from economy, through mid range, to full performance ratio. This requirement is probably the hardest to obtain in practice, if all of the " Idealomatic's " other features are to be retained.

The ideal of absolute smoothness exists in the fluid torque converter type of transmission and it is the objective of every friction clutch enthusiast to produce automatic shifts that would equal its smoothness and response.

Specifically, the advantages that the converter holds over the geared ratio transmission in this respect are its ability to change ratio without a cessation of drive line torque and its direct and immediate response to a change of engine torque. Friction clutch shifts require that drive line torque be broken to change ratio and response must be provided by an outside control that attempts to sense engine torque change.

The problems inherent in producing a simple, low cost control, free from manufacturing variations and with an absolutely smooth shift, are numerous and, to date, unsolved. As a result, some manufacturers have compromised other factors considerably to provide the converter's smoothness in the normal driving range.

COST OF THE TRANSMISSION

Cost is the fourth major item that requires compromise in today's passenger car transmissions. The " Idealomatic " would, of course, perform all of the functions previously mentioned, as well as the miscellaneous features given below, so simply and cheaply that it could be priced competitively with present transmissions.

OVERRUN BRAKING

The requirement that the transmission provide braking ability for coasting downhill has caused additional compromise in automatic transmission design. It is apparent that one gear ratio will not provide the most pleasing braking ability for all car speeds and grades. A recently announced innovation in automatic transmissions provides three overrun braking ranges, manually selectable by the driver. The " Idealomatic," therefore, must have at least this number and, to be truly ideal, must provide an infinitely variable overrun ratio, sensitive to both grade and car speed.

Converters are useless with respect to providing engine friction braking without the addition of mechanically geared ratios. Owing to the free wheelers required in their torque multiplying members, converters furnish nothing but an inefficient fluid coupling drive when coasting. Introduction of a gearset for braking only in a converter design is a costly compromise.

Geared ratio transmissions, on the other hand, provide cheaply the same gear ratio and clutched drive used in forward gear for braking application.

NOISE LEVEL

The " Idealomatic " must be noiseless in operation. Freedom from converter cavitation, whine, gear noise, hydraulic pump noise, etc., would be essential.

It should be apparent from the foregoing that the " Idealomatic " will require considerable development of, and probably deviation from, the fundamental designs of today, when it is produced.

HUNGARY

Stress-Strain Relation of Rubber Blocks Under Compression

By G. SCHAY and P. SZÖR. (From *Acta Chimica*, Vol. 2, No. 4, 1952, pp. 317-329, 7 illustrations.)

For the stress-strain relation of differently shaped rubber blocks submitted to compression, an equation of general applicability has been deduced, starting from the idea that compression work must also be done against the tension arising through the increase of the free surface. The validity of the equation obtained was tested by measurements on cylinders as well as by data on quadratic prisms published in previous literature. The calculated values are in good agreement with the experimental data, within the limits of experimental error.

ONE of the prerequisites of correct dimensioning of rubber springs is an exact knowledge of the stress-strain relation. Rubber springs are liable to be subjected to compression, shear and torsion. It has been found that the stress-strain curve is, in general, greatly dependent on the shape of the rubber spring, but independent of its absolute dimensions, so that geometrically similar bodies behave in the same manner. In the present paper it is intended to deal with compression only. Until now, only a limited series of experiments and tabulated values derived therefrom is available as to the shape of the stress-strain curve[1],whereas their theoretical co-ordination has not yet been made. In the following, an equation is derived by which these curves can be calculated on a uniform basis, provided the shear modulus of the rubber substance and only one experimental constant, which depends solely on the shape of the rubber block and which may be determined by a single measurement, are known.

The only case of practical interest is when compression of the rubber block is effected by a stress acting upon two parallel faces of the block, these faces being fixed to the solid surfaces in contact with them (metal fittings) in such a way that no slip can occur. Under these conditions, these surfaces remain unchanged at any degree of compression. The deformation of the block, consequently, manifests itself only at the free sides and, as experience shows, is always

accompanied by an increase of the area of these surfaces, at least in the case of block-shapes and strains of practical interest. The reason for this increase lies in the fact that the volume of the rubber remains practically unchanged even when compressed to the greatest extent occurring in practice[2], i.e., the block gives way to the applied stress only by a change of its form. This is one of the main differences between the elastic behaviour of rubber and that of normal elastic solids. The other important difference is that, with normal solids, the yield point confines the range of elastic deformations to very small values, whereas rubber blocks can be compressed in a completely elastic manner to a relatively small fraction of their original height. Consequently, *Hooke's* proportionality law cannot be applied in the case of rubber, the stress increasing more rapidly than the deformation.

The statistical theory of rubber-elasticity is based on the fact that an unstrained rubber block is a statistically isotropic network of segments of chain molecules held together by randomly distributed junctions. The theory supposes that all volume elements which have a shape similar to that of the whole body undergo a deformation proportional to that of the whole. In the case of compression, it would be acceptable, only if the faces of the rubber block in contact with the pressing surfaces were permitted to glide freely thereon (in this case, e.g., a compressed cylinder would presumably remain cylindrical). In practice, however, only completely fixed surfaces are dealt with, and these naturally remain unstrained. The surface network is forced apart in this way by the lateral elongation of the interior of the block. This means that there must be a definite relation between the shear stress in the interior of the block and the surface stress on the other part.

From this relation between the stresses arising on compression, it should be possible in principle to calculate, not only the shape adopted on deformation of the blocks, but the relation between the pressure acting on the fixed surfaces and the change in height, i.e., the stress-strain function as well. Actually, however, the problem cannot as yet be solved in this fashion. In the subsequent treatment, an approximate procedure was chosen, introducing a correction term beside the term expressing the elastic resistance of the main body of the block and this latter term may be taken as identical with the usual statistical expression. The new correction term contains the stress surplus in the surface coming from the lateral strains, as determined empirically by the measured surface increase.

The relation between stress p (the perpendicular force acting upon the unit of bound surface area) and deformation λ (the ratio of the block's actual and original heights) may be derived from the expression of the work of deformation, the differential of this per unit volume of the block being:

$$dW = p d\lambda \quad .. \quad .. \quad .. \quad (1)$$

According to the above assumptions, the work dW is composed of two parts. One of these, the work done against the elastic resistance of the whole of the block, can be expressed by:

$$d_1 W = Gf(\lambda)\, d\lambda \quad .. \quad .. \quad (2)$$

where, according to the statistical theory of rubber elasticity,

$$f(\lambda) = \lambda - 1/\lambda^2 \quad .. \quad .. \quad (3)$$

and G is the shear modulus of the rubber substance.

In order to find an explicit expression for the part of the whole work associated with the increase of the free surface area, let it be assumed that the stress σ striving to bring the surface back to its original dimensions can be put in the form:

$$= Hg(\alpha) \quad .. \quad .. \quad (4)$$

where H is some characteristic constant and g some function of the relative surface α awaiting explicit expression. At the moment, it can be stated only that this function must disappear when $\alpha = 1$ and must be otherwise a monotonic increasing function of α. The work done when increasing the whole of the free surface F by a relative increment $d\alpha$ follows from eq. (4)

$$dL = FHg(\alpha)\, d\alpha \quad .. \quad .. \quad (5)$$

As the total work in eq. (1) has been referred to the unit volume of the block, the surface work eq. (5) must be expressed in the same way, giving:

$$d_2 W = \frac{dL}{V} = \frac{F}{V} Hg(\alpha)\, d\alpha \quad .. \quad (6)$$

where V is the volume of the block.

The experimental fact that the stress-strain curves of blocks having the same shape are independent of their absolute dimensions justifies the assumption that, for a given shape, α must be a unique function of λ only, that is:

$$d\alpha = \frac{d\alpha}{d\lambda} d\lambda \quad .. \quad .. \quad .. \quad (7)$$

On the assumption that:

$$dW = d_1 W + d_2 W \quad .. \quad .. \quad (8)$$

the stress p can be expressed from eq. (1):

$$p = \frac{dW}{d\lambda} = Gf(\lambda) + \frac{F}{V} Hg(\alpha) \frac{d\alpha}{d\lambda} \quad .. \quad (9)$$

Here the right hand side is, apart from the constant characteristic of the rubber substance, a function of the deformation, but its second term is a function of the shape of the rubber block as well. Hence eq. (9) is the sought equation of the stress-strain curve, which may be, however, of use only if explicit forms can be found for the functions $Hg(\alpha)$ and $\alpha = \alpha(\lambda)$.

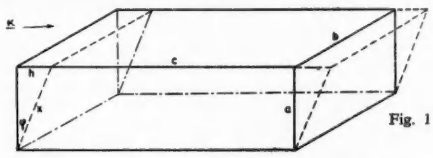

Fig. 1

To determine the form of the first of the above functions, it should be considered that all intermediate layers of the rubber subjected to compression undergo shear deformation parallel to the surfaces to which the pressure is applied, this shear leading to a bulging of the free lateral faces. Hence, there must be an intimate relationship between the shear and the tension of the surface. This relationship may be demonstrated on the following model: Consider a prism with edges a, b and c (Fig. 1) which undergoes an elastic shear deformation by the force K acting in the direction and at the place shown in the figure. For not too large values of the angle of deformation:

$$K = G\, bc \tan \varphi = G\, bc\, h/a \quad .. \quad (10)$$

where G is the shear modulus. It is obvious, however, that the front of the prism undergoes a simultaneous increase from ab to xb, and that the surface tension induced thereby is given by the part of the component of force K lying in the plane xb and perpendicular to the edge b, referred to unit length of edge b, i.e.:

$$\sigma = \frac{K}{b} \sin \varphi = \frac{K}{b} \frac{h}{x} \quad .. \quad .. \quad (11)$$

When expressing K from eq. (10), and making use of the expression (4) of σ, and considering that the free surface of the prism is $F = ab$ and its volume $V = abc$, it follows that:

$$\frac{F}{V} Hg(\alpha) = G \frac{h^2}{ax} \quad .. \quad .. \quad (12)$$

It is also apparent from the figure that the relative increase of the surface is

$$\alpha = \frac{xb}{ab} = \frac{x}{a} \quad .. \quad .. \quad (13)$$

Furthermore, $h^2 = x^2 - a^2$. With these equations, eq. (12) may be further transformed:

$$\frac{F}{V} Hg(\alpha) = G \frac{x^2 - a^2}{ax} = G \left[\frac{x}{a} - \frac{a}{x} \right] = G \left[\alpha - \frac{1}{\alpha} \right] \quad (14)$$

As eq. (14) no longer contains the dimensional characteristic of the prism in question and, moreover, any rubber block can be built of rectangular prisms to any desired degree of accuracy, it seems a safe assumption that this relationship is generally applicable. As a result of the foregoing, eq. (9) becomes:

$$p = G \left[\left(\lambda - \frac{1}{\lambda^2} \right) + \left(\alpha - \frac{1}{\alpha} \right) \frac{d\alpha}{d\lambda} \right] \quad .. \quad (15)$$

The function $\alpha(\lambda)$, which is still unknown, cannot be given a general form since it depends on the shape of the rubber block. At present, experiments and calculations are based on rubber blocks having either a cylindrical or a square prismatic form only.

I. Cylindrical Blocks

Let the radius of the cylinder be r and its height m. On compression, this height decreases to $m\lambda$. Fig. 2a shows a cylinder where $m = 2r$, and in order to have a better idea of the deformations occurring on compression, a network is depicted on the mantle of the cylinder. Fig. 2b shows the same cylinder under compression. It is seen that the middle part (about 3/5th) of the compressed cylinder remains approximately cylindrical, though even here the generatrices show a slight barrel-like bulge. At the lower and upper parts, however, the generatrices join the pressing surfaces with a strong curvature, and the cylinder leans against these surfaces in close contact with them, at a greater width than in its original state. No simple function can be found to describe the deformation of the generatrices precisely, and therefore an approximation of the actual deformation has to be used for the calculation of the surface increase.

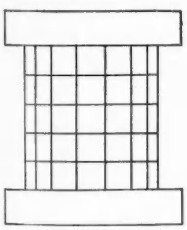

Fig. 2a

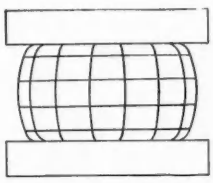

Fig. 2b

The slight barrel-like bulge in the middle may be considered as being eliminated by means of some squeezing ring. By this procedure, this part is transformed to a real cylinder of radius ρ. Then the rubber evading this squeeze must flow into the lower and upper parts, increasing simultaneously the joining curvatures of the generatrix. The curve obtained in this way is taken to be the quarter of a circle, with radius d. Visibly, this additional imaginary deformation does not change the surface area by any considerable amount. This simplified picture of the actual deformation is shown in Fig. 3.

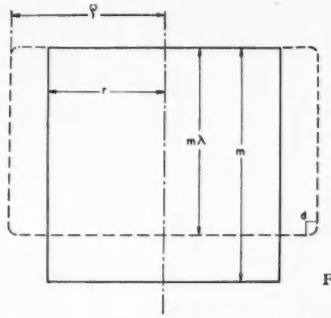

Fig. 3

The volume of the cylinder when unloaded is:

$$V = \pi r^2 m \quad .. \quad .. \quad .. \quad (16)$$

whereas that of the deformed body as shown in Fig. 3 is:

$$V = \pi \rho^2 m\lambda - \pi [(4 - \pi)(\rho - d) d^2 + \tfrac{2}{3} d^3] \quad .. \quad (17)$$

As d is always relatively small, the second term, containing this quantity only in higher powers, is a practically negligible correction. Since the volume of the rubber block remains unchanged on compression, on comparison of the right hand sides of (16) and (17)

$$r^2 \approx \rho^2 \lambda \quad .. \quad .. \quad (18)$$

is obtained, or:

$$\rho = r/\sqrt{\lambda} \quad .. \quad .. \quad (19)$$

The next task is to compute the increase of the free surface. The free surface of the original cylinder is made up only of its mantle, that is

$$F(1) = 2\pi r m \quad .. \quad .. \quad (20)$$

Taking the part of the original cylinder's mantle now in close contact with the pressing surfaces as "free" surface as well, the free surface of the deformed body, as shown in Fig. 3, however, is given by

$$F(\lambda) = 2\pi \rho m\lambda + 2\pi(\rho^2 - r^2) - 2\pi [\rho d(4 - \pi) + d^2(\pi - 3)] \quad (21)$$

Part of the last term, owing to the rounding off of the edges, now contains the first power of d and this cannot be neglected (the term of the second degree in d, however, may be neglected in this case).

Observations made on cylinders compressed to various extents have led to the assumption that the radius of the curvature d can be taken proportional to the increase of the radius of the cylinder's mantle, that is

$$d = k(\rho - r) \quad .. \quad .. \quad (22)$$

The increase of the free surface is evidently:

$$\alpha(\lambda) = F(\lambda)/F(1) \quad .. \quad .. \quad (23)$$

Substituting into the expression (21) of $F(\lambda)$ the value (22) of d and that of ρ according to eq. (19), and neglecting the term of the second degree with respect to d, gives

$$\alpha(\lambda) = \sqrt{\lambda} + \frac{r}{m}\left(\frac{1}{\lambda} - 1\right) - k\frac{r}{m}(4 - \pi)\left(\frac{1}{\lambda} - \frac{1}{\sqrt{\lambda}}\right) \quad (24)$$

The quotient r/m in this equation is a value characteristic of the shape of the cylinder, giving the ratio of free and fixed surfaces, which can also be called "stockiness" index. This index is generally used to characterise the shape of rubber springs. From eq. (24) it follows that the relative surface increase is a function not only of the relative compression λ but of the shape as well, a fact emphasised by writing:

$$\alpha(\lambda, \delta) = \sqrt{\lambda} + \delta\left(\frac{1}{\lambda} - 1\right) - k\delta(4 - \pi)\left(\frac{1}{\lambda} - \frac{1}{\sqrt{\lambda}}\right) \quad (25)$$

where $\delta = r/m$ is the above mentioned index of "stockiness." The differential quotient occurring in the basic equation (15) is obtained by differentiating eq. (25):

$$\frac{\partial\alpha(\lambda,\delta)}{\partial\lambda} = \frac{1}{2\sqrt{\lambda}} - \frac{\delta}{\lambda^2} + k\delta(4-\pi)\left(\frac{1}{\lambda^2} - \frac{1}{2\lambda\sqrt{\lambda}}\right) \quad (26)$$

It should be noted that, with the values of k to be determined in the following (0·57 with a cylindrical body, 0·46 with a quadratic prism), in the case of slenderer bodies (with cylinders $\alpha < 0{\cdot}662$, with prisms $\alpha < 0{\cdot}623$) the values of α according to eq. (25) show a very little decrease at the start alongside with λ. The minimum values of α are 0·987 and 0·992 respectively when $\delta = 0{\cdot}5$. On greater compressions, however, α increases rather rapidly even in this case. This behaviour of the function does not mean that with slenderer bodies the surface should actually decrease at the start of the deformation, but only that the idealised picture of the deformation as shown in Fig. 3 is not a satisfactory approximation in this range. Hence, with the value $\delta = 0{\cdot}5$, the value $\alpha = 1$ was chosen in the range where, according to eq. (25), $\alpha < 1$ would have been obtained. This is equivalent to neglecting the correction term in eq. (15); in other words, with blocks of such a shape (not usual in spring technique) the usual statistical expression of rubber elasticity in the range of small compressions was applied. The results so obtained were entirely satisfactory.

Substituting the expressions thus obtained into eq. (15), this equation becomes suitable for the calculation of the stress-strain curves, assuming that the shear modulus G and the numerical factor k are known. The latter was determined according to eq. (22), measurements taken from the photographs on which Figs. 2a and 2b are based, yielding

$r = 33{\cdot}7$ mm $\rho = 40{\cdot}8$ mm and $d = 4{\cdot}05$ mm and hence:

$$k = d/(\rho - r) = 0{\cdot}57$$

To test the validity of the expression obtained, rubber cylinders of different "stockiness" (but having equal bases of 1 cm²) were made and their stress-strain curves determined in Höppler's consistometer at 25 C. The amount of strain was read ten minutes after the stress had been applied, in order to eliminate the effect of a slight initial relaxation. As the base and upper faces can hardly be obtained strictly parallel in the process of vulcanisation, the starting point of the stress-strain curve had to be established from the curve itself by extrapolation. The composition of the rubber-mixture used was as follows:

Rubber	.. ..	100·0 parts
Sulphur	.. ..	2·8 ,,
Carbon black (soft)	..	20·0 ,,
Mercaptobenzothiazole		0·8 ,,
Antioxidant	.. ..	0·3 ,,
Stearic acid	.. ..	3·7 ,,
Pine tar	.. ..	2·2 ,,
Zinc oxide	.. ..	5·0 ,,

The value of the shear modulus G was found to be 9 kg/cm² from the measurements made on a rubber cylinder with index $\delta = 0{\cdot}5$.

The experimental results are shown in Fig. 4. Here the lines represent the curves calculated from eq. (15) for the corresponding values of δ indicated in the figure, using the values $G = 9$ kg/cm² and $k = 0{\cdot}57$. The marks represent measured data. It can be seen that the calculation yields values in close agreement with the experimental data within the limits of error of the latter. It should be noted that these limits are, especially in case of small deformations, relatively great.

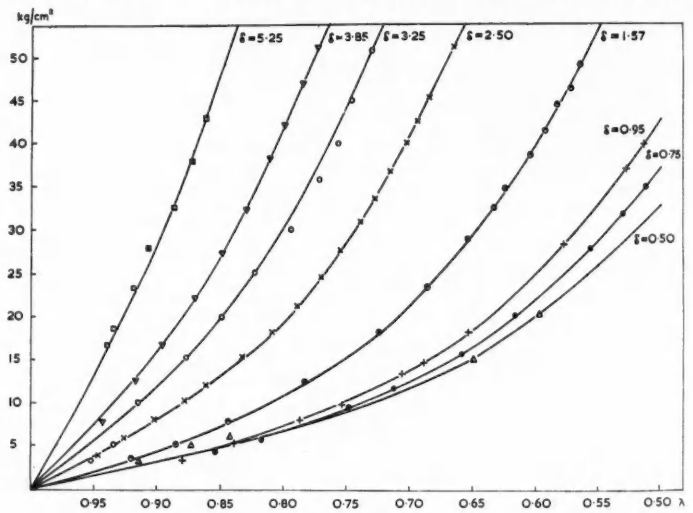

Fig. 4

II. Quadratic Prisms

Similar tests on deformation under compression were carried out with rubber cubes. Results showed that in this case too, considerable parts of the free edges of the cube remain straight, i.e., the edges of a cube undergo a deformation similar to that of the generatrices of a cylinder. Hence, the problem of the actual deformation can be approached in the same way, so that Fig. 3 can also be regarded as the idealised section of a deformed quadratic prism, the only difference being that the diameter $2r$ of the cylinder must be replaced by the edge of the base of the prism. The calculation of the volume and the surface made by this substitution yields the same result already obtained for a cylinder, i.e., eqs. (25) and (26) retain their validity even in this case.

No detailed measurements were made with quadratic prisms, as the measurements of Jenkins[2] were available to test the validity of the equations deduced.

The differences between values calculated from eq. (15) and Jenkins' experimental data were found to be small, not exceeding 10 per cent, even with a threefold increase of the surface.

The measurements of Jenkins also extended to cylindrical bodies. Without giving detailed information, these state that stresses measured on cylinders are in general smaller by a factor of 0·9 than those obtained with quadratic prisms with identical values of δ. In the equation deduced here, the difference between the two different bodies appears only in the value of the factor k (0·57 with a cylinder and 0·46 with a quadratic prism). With these values of k, the present equation yields values in close agreement with the data of Jenkins in this respect as well.

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Solid Fuel for Internal Combustion Engines

By J. S. Moloney. (From *The Engineering Journal*, Vol. 36, No. 4, April, 1953, pp. 395-401, 9 illustrations.)

This paper deals with the transport of solid particles of coal suspended in air, and the use of pulverised coal as a fuel in internal combustion engines. The relation between the velocity of a coal particle in still air and the resulting frictional resistance force has been found experimentally for different coal particle sizes. Test results indicate some of the factors to be considered in engine design and operation. A comparison is made between values of drag coefficient for coal particles moving in air, and those for spheres, at the same Reynolds number.

The operation of internal combustion engines with powdered fuel is similar to that of the diesel engine, and has the advantage of relative cheapness of fuel. For instance, at the present time, fuel cost per unit of energy output for a coal dust engine (assuming 30 per cent efficiency) would be roughly one-fifth that for a petrol engine (at 20 per cent efficiency), two-fifths that for a diesel engine (at 30 per cent efficiency), or one-fifth that for a steam engine using coal (at 6 per cent efficiency).

Design Considerations

The injection of the solid particles into the engine, the formation of a combustible mixture by thorough mixing with the correct quantity of air and the expulsion of the combustible residue all require the dust to be fluid-borne in the air, the air acting as a carrier. The more practical problems of increased wear and lubricating oil consumption, inherent in this type of engine, depend largely on the extent to which the solid is prevented from making contact with the working surfaces by remaining fluid-borne. The maximum allowable fuel particle size in any particular engine for satisfactory operation depends on the extent of the permissible relative movement between the solid particles and the fluid in which they are carried, and this, in turn, depends on both the engine design and operating conditions.

For laboratory measurement of the rate of settling of coal dust in air, a simple form of electrostatic precipitator was employed, as shown in Fig. 1. The electrodes were placed close together to give as high a value of potential gradient as was practicable without sparking. The shiny aluminium electrode used provided a surface that was suitable for observing faint dust deposits. The intermittent high voltage from one of the spark leads of a magneto was used as a voltage supply.

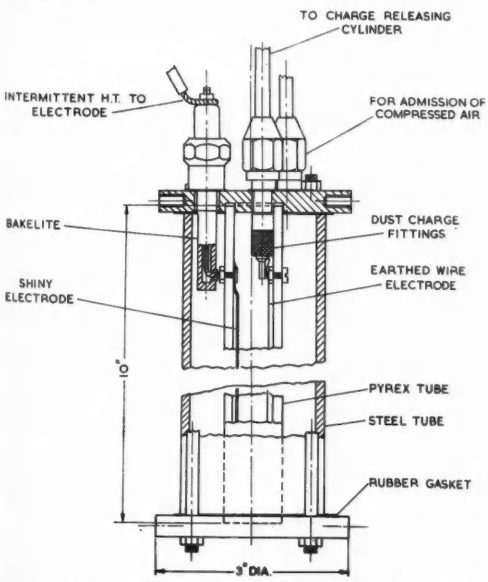

Fig. 1. **Apparatus for measuring settling velocity of coal dust particles in air at various pressures.**

For measuring the settling velocity of dust particles at accelerations greater than that of gravity, a centrifuge was used (Fig. 2). The angle of the dust precipitator tube was adjustable so that its axis could be set in line with the mean direction of travel of the dust particles, the actual particle path not being a straight line relative to the centrifuge table in either the horizontal or the vertical plane. Calculations of centrifugal acceleration were made for approximately the mid-point of the precipitator tube, i.e., using a centrifugal radius of 2·5 ft.

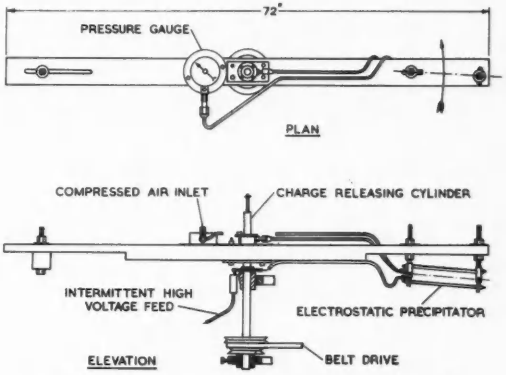

Fig. 2. **Centrifuge for measuring coal dust settling velocity with acceleration greater than that of gravity.**

The dust to be tested was kept at a temperature of about 70° C to ensure dryness. The shiny aluminium strip electrode was prepared for the test by detaching it from the precipitator tube, wiping it clean, applying uniformly a thin coat of grease, and replacing it in the tube. The small dust charge capsule was then charged one-third full (0·015 cc approximately) of the dust to be tested, placed in the capsule nut, and the latter screwed into position (see Fig. 1). The precipitator tube was then set in place, the centrifuge was started, and the magneto for intermittent voltage supply turned on. The centrifuge and magneto speeds were recorded and the charge of dust was released from its receptacle in the upper end of the precipitator tube by injecting a small blast of air from the charge-releasing cylinder (see Fig. 2). Release of the dust charge took place when one side of the thin rubber flapper valve in the dust charge capsule was forced through the opening on the tiny end plate to which it was cemented at one point. For tests at pressures higher than atmospheric, compressed air was admitted to the dust precipitator tube. After stopping the centrifuge, the shiny electrode was removed from the dust precipitator. The dust could be observed deposited in waves along the electrode. Some experience was required in learning how to measure the distance between the deposits before results could be repeated. Apparently, the dust cloud

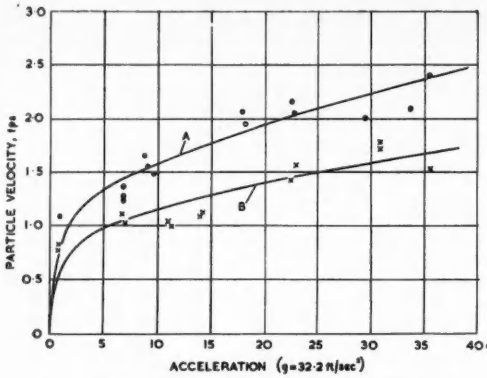

Fig. 3. **Results of coal dust particle settling tests.** Particle sizes = 0·084 mm (curve *A*) and 0·064 mm (curve *B*).

consisted of several small clusters, giving deposits in separate series of waves for each cluster.

The settling velocity was found by taking the product of the distance between the deposits and the frequency of the voltage pulses.

Bituminous coal was used in these tests. A mean diameter of a representative particle from each graded sample of dust, measured under the microscope, was taken as the particle size of the sample. Results for four different particle sizes (0·084 mm to 0·046 mm) are plotted in Figs. 3 and 4.

Discussion of Results

In the velocity-acceleration curves, the force acting on a particle is equal to the product of particle mass and acceleration. It is also equal to the force of frictional resistance between the particle and the carrier fluid. Assuming that this force varies as some power of velocity, the equation may be written :

$$F = Mg = kV^a \text{ or } g = KV^a$$

where F = force ; M = mass of particle ; g = acceleration ; V = velocity ; and K, k, and a are constants.

It has been shown* that, for an obstacle immersed in a flowing medium, the drag coefficient C_D is a function of Reynolds number R only. Note that drag force $F = C_D \rho D^2 (V^2/2) (\pi/4)$ where ρ = density of fluid, and D = diameter in the case of spheres. A graph of C_D as a function of R given for flow of a stream past a sphere, shows that

$$C_D = 250, \text{ when } R = 0{\cdot}1$$
$$C_D = 30, \text{ when } R = 1$$
$$\text{and } C_D = 4{\cdot}5, \text{ when } R = 10$$

These values of R are of the order of those encountered in the tests.

Values of C_D and R for the coal dust settling-tests have been calculated and are plotted in Fig. 5 in the same graph with the $C_D - R$ curve for spheres. C_D increases as R increases, instead of decreasing as it does for spheres. It is interesting to note in comparing the $C_D - R$ curve for rough pipes† and the $C_D - R$ curve for discs with that for spheres*, that the change in slope is in the same direction when going from rough flow conditions to smooth.

It is evident from Fig. 5 that the smaller the particles become, the greater is the relative effect of their surface roughness for a given value of R. Presumably, this is because surface phenomena of small objects increase with decrease in size.

*S. Goldstein : Ed. Clarendon Press, 1938, Vol. 1, pp. 13-16. Modern Developments in Fluid Dynamics.

†C. F. Colbrook : Inst. C.E., G.11, Feb. 1939, p. 133. Turbulent Flow in Pipes, with particular reference to the Transition Region between the Smooth and Rough Pipe Laws.

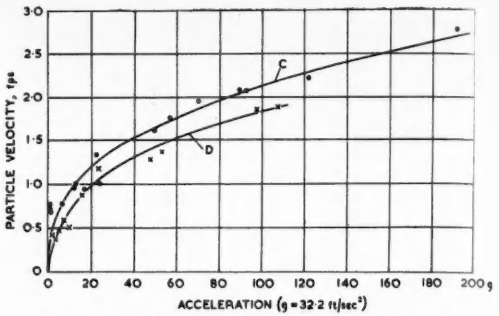

Fig. 4. **Results of coal dust particle settling tests.** Particle sizes = 0·052 mm (curve *C*) and 0·046 mm (curve *D*).

At very low values of R for each particle size used, the $C_D - R$ curves for the coal dust particles appear to become tangent to a line parallel to the $C_D - R$ curve for spheres. If these curves do, in fact, become the same as the curve for spheres, when R becomes very small, then the effects of roughness are nil at very low R. Error in measuring mean particle diameter would account for the separation of this tangent line —dotted in Fig. 5—from the $C_D - R$ curve for spheres. It seems likely that, at this condition, flow would be completely laminar, since friction would be the same for both spheres and coal particles and would be due entirely to fluid viscosity μ and not to fluid density ρ.

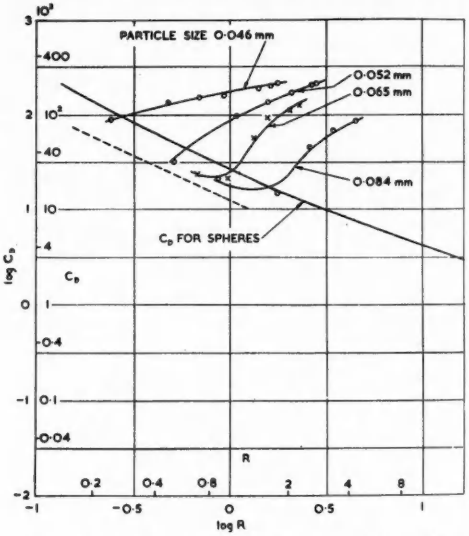

Fig. 5. $C_D - R$ **values from tests, plotted with the** $C_D - R$ **curve for spheres.**

Application of Results

In a flowing mixture of solid particles and fluid, consider the motion of a small particle of fluid which contains a much smaller solid particle. Assuming that absolute velocity is much greater than relative velocity between solid and fluid, it may be said that, if absolute acceleration remains constant, relative velocity also remains constant.

If a mixture of solid particles in fluid is to move a given distance with the least amount of relative motion between solid and fluid, absolute acceleration should be high up to the running velocity, and then it should

drop to zero. Thus, final running velocity is attained quickly and then remains constant, i.e., its maximum value in practice must be kept as low as circumstances permit. In bringing the mixture to rest, the order must be reversed if the same effect is to be obtained.

The tests with coal dust establish that, when absolute velocity of a flow system is increased, relative motion between dust particles and carrier fluid is decreased. Thus, for example, if the speed of a dust engine is increased, the chances of fuel clogging and the extent of abrasion of working surfaces are decreased. This refers, of course, to abrasion other than that due to particle impact, such as might occur in turbine blading.

It also follows from the above that, in the design of an internal combustion engine for solid fuel, or in any flow system where dust is carried in a gas stream, the points in the flow circuit of the working fluid where velocity is at a maximum should be designed to ensure that this maximum value is as low as possible.

When the maximum limit of fuel particle size has been found for a particular set of conditions by engine tests, a reasonably intelligent estimate of the maximum allowable particle size can be made for any other engine and any other operating condition, by use of the appropriate velocity curves for the complete engine cycle.

Summary and Conclusions

Coal particle velocity relative to the air in which the particle is immersed has been found for various values of acceleration of coal-air mixtures for several different coal particle sizes. Results given in this paper are for particle sizes from 0·084 mm to 0·046 mm mean diameter. The maximum value of acceleration used was 191 times that of gravity.

The value of the drag coefficient C_D, which depends only on Reynolds number R, for geometrically similar objects immersed in a flowing fluid, was found to increase with R for dust particles over the range of R encountered during the tests (i.e., R between 0·25 and 8 approximately), whereas C_D increases with increase in R for spheres. This difference is attributed to the difference between the rough surface of the coal particles and the smooth surface of a sphere.

It seems very likely that problems of wear and clogging due to fuel particles in a solid fuel engine would become less important as engine speed increases, though this requires confirmation by actual engine tests.

It is not unlikely that solid fuel for internal combustion engines will find useful application in the future, allowing for reasonable development and considering the internal combustion engine to take the form of a combustion gas turbine for large transport applications, e.g., for locomotives or ships. This possibility becomes greater if reports concerning the limited world supply of liquid fuels are to be taken seriously. The long-term prospects of using solid fuel thus depend simply on engineering economics. Solid fuels other than coal might, of course, be used.

GERMANY

Investigation into Erosion of Gear Teeth

By G. Niemann and H. Glaubitz. (From *VDI-Zeitschrift*, Vol. 95, No. 6, February 21, 1953, pp. 164-166, 12 illustrations.)

Tests on hardened spur gears under increasing loads were undertaken to clarify the following questions:—

(a) Whether erosion phenomena are reproducible,

(b) whether pronounced unilateral modification of the profile, i.e., higher velocity of sliding at the tooth face, decreases the critical load at which erosion starts,

(c) how far additives to oil increase the critical load,

(d) whether gears, run in with hypoid oil, can sustain a higher critical load than gears run entirely on mineral oil.

Straight spur gears made of steel with 0·17 to 0·22 C, 1·1 to 1·4 Mn, 1·0 to 1·3 Cr, were case-hardened and ground by the Maag cross-grinding method; the tips were not radiused. The dimensions (see Table) show

Table of dimensions and running characteristics of modified straight spur gears.

Dimensions of teeth		Pinion	Wheel
Pressure angle		20°	20°
Number of teeth		16	24
Module	mm	4·5	4·5
Diametral Pitch		5·644	5·644
Pitch Diameter	mm	72	108
	in.	2·835	4·252
Outside Diameter	mm	88·77	112·50
	in.	3·495	4·429
Root Diameter	mm	68·97	92·70
	in.	2·715	3·650
Factor of modification		+0·8635	−0·5
Face width	mm	20	20
	in.	0·787	0·787
Running Characteristics			
Distance between centres	mm	91·5	
	in.	3·602	
Actual pressure angle		22° 20′	20° 20′
Dia. of meshing circle	mm	73·2	109·8
	in.	2·882	4·323
Running speed	rpm	2175	1450
Circumferential speed v at meshing circle	fps	27·33	27·33
Velocity of sliding at tip of tooth		1·022 v	0·162 v
Specific sliding at tip of tooth	%	+87	+37
Specific sliding at working depth	%	−57	−650

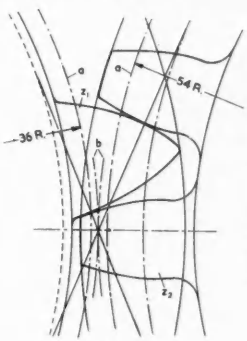

Fig. 1

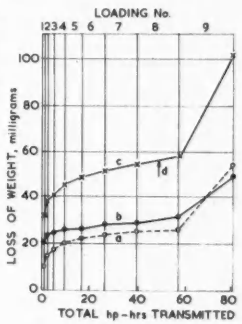

Fig. 2

Fig. 1. **Modified tooth form of spur gears under investigation.**
a = Nominal pitch circles for generation of gears, b = meshing circles, z_1 = pinion, z_2 = wheel.

Fig. 2. **Curves showing wear of modified spur gears run with gear oil.**
a = Pinion, b = wheel, c = pinion and wheel, d = start of erosion. No. of loading: 1 = 20·8 lbs, 4 = 370 lb, 5 = 575 lb, 6 = 820 lb, 7 = 1065 lb, 8 = 1475 lb, 9 = 1890 lb total tangential tooth load.

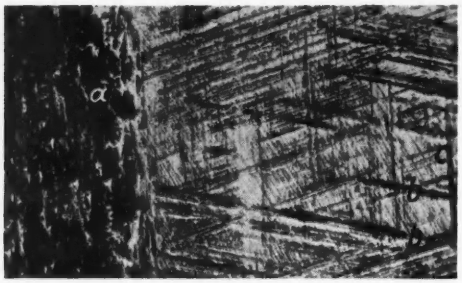

Fig. 3. **Erosion after running with mineral oil.** Magn. × 80. *a* = Erosion, *b* = grinding scratches, *c* = abrasions.

that modification of the profile produces relatively high sliding and places the meshing circle near the tip of the wheel and the root of the pinion tooth (Fig. 1.).

The teeth were lubricated by dipping in oil, the oil level reaching up to the root of the wheel teeth. The oil bath was heated electrically to 80° C while the gears were idling. The heating was then switched off and the load applied. After ½ hour the oil temperature was measured and the gears were stopped, dismantled, and inspected for erosion. The loss of weight due to wear was measured by weighing. The test was repeated with the next higher load, and so on, until the first traces of erosion became visible. In some cases an additional test run with a still higher loading was carried out. The oils used were (1) pure mineral oil, (2) gear oil (high-pressure mineral oil, slightly doped), (3) hypoid oil (extreme-pressure mineral oil, heavily doped). All the above oils were similar in viscosity to SAE 90.

Fig. 2 shows a typical test curve. Wear first increases strongly during the running-in period (loadings 1 to 4), then remains fairly steady (loadings 5 to 8). At *d* (loading 8) the first traces of erosion appear, and continued erosion (loading 9) leads to a heavy increase in wear. With pure mineral oil, erosion was first observed after loading 6 (tangential tooth load = 820 lb); with gear oil it occurred after loading 8 (1475 lb); with hypoid oil after loading 11 (3825 lb). When mineral oil was used after running in with hypoid oil, erosion started after loading 7 (1065 lb). The measured temperatures of the oil bath remained about the same for all three oils.

The micro-structure of the eroded part of the face appeared the same whichever oil was used (Fig. 3). A shallow depression is seen lacking the characteristic groove formation of simple abrasion. The profile roughness of an erosion also differs characteristically from that of an abrasion and is mainly independent of the oil used (Fig. 4). The distribution of erosion over the face width, however, depends markedly on the oil used: fairly wide eroded faces occur at once with mineral oil, whereas with gear oil and a higher loading the erosion is narrower, and with hypoid oil and the highest load it is in quite narrow strips, often very close together (Figs. 5 to 7). Where erosion has appeared,

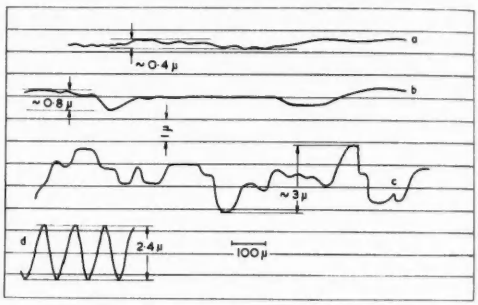

Fig. 4. **Surface roughness of a tooth face measured in the direction of the gear width.** *a* = Ground face before test, *b* = abrasion scratches, *c* = erosion, *d* = roughness measured with comparator.

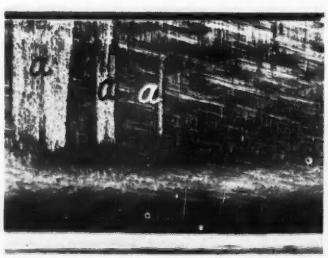

Fig. 5. **Erosion after running with mineral oil at 820 lb.**

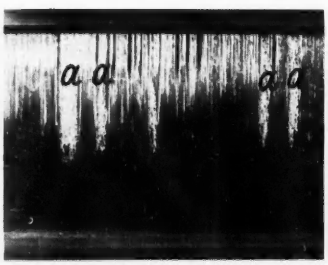

Fig. 6. **Erosion after running with gear oil at 1890 lb.**

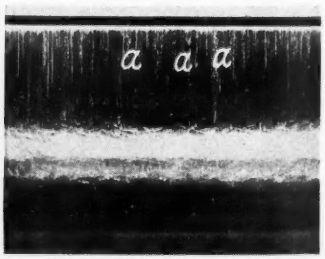

Fig. 7. **Erosion after running with hypoid oil at 2825 lb.**

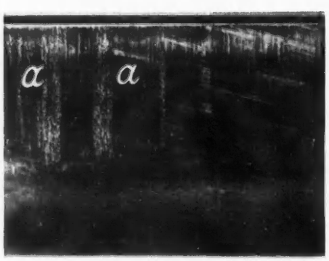

Fig. 8. **Erosion of Fig. 5 is smoothed out after running with hypoid oil at 1065 lb.**

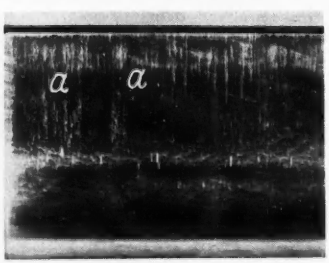

Fig. 9. **Erosion of Figs. 5 and 8 has nearly disappeared after running with hypoid at loads increasing from 1475 to 2825 lb.**

Figs. 5, 6, 7, 8 and 9. **Traces of erosion *a* on the tooth face of modified pinion after running with various oils at different tangential tooth loads.**

Magn. × 3 × ⅔

it can be progressively smoothed down and even eliminated by the use of hypoid oil (Figs. 8 and 9). If, after treatment with hypoid oil, the gears are run again with mineral oil, new erosion begins to appear at a higher load than without the treatment (1065 instead 820 lb tooth load).

Summary

(a) The first traces of erosion always occur at the same loading if the same oil is used and running conditions are kept constant.

(b) The influence of the modification of the profile, i.e., of the velocity of sliding, is considerable. With standard teeth and gear oil, a tangential tooth load of 3340 lb does not produce any signs of erosion while modified teeth erode at a load of only 1475 lb.

(c) The critical load varies widely with the type of oil used and can be raised appreciably by the use of hypoid oil. Modified gears begin to erode at a tangential tooth load of 820 lb with pure mineral oil, 1475 lb with gear oil, 2825 lb with hypoid oil.

(d) Erosion produced by running with mineral oil can be smoothed out if hypoid oil is used at the same loading. Running-in with hypoid oil also increases the critical load at which erosion sets in, when the gears are subsequently run with mineral oil. Hypoid oils are therefore recommended for the running-in period of gears and are probably equally suitable for sliding bearings.

U. S. A.

The Aerodynamic Profile as Acoustic Noise Generator

By M. KRAMER. (From *Journal of the Aeronautical Sciences*, Vol. 20, No. 4, April, 1953, pp. 280-282, 296, 1 illustration.)

Explanations of the origin of profile noise have often been attempted. However, even the most recent explanation fails to make understandable the outstanding role played by the Mach Number. This paper gives a recent explanation of profile noise, reports some new experiments and states the author's opinion of its predominating source.

A RECENT explanation[1] of the origin of the generation of profile noise is as follows: "The good profile radiates a broader frequency band (than the poor one). As explanation, we might assume that the vortices (of the vortex stream leaving the profile) are generated in a kind of feedback mechanism. A vortex leaving the profile produces a disturbance in the air which travels with sound velocity to the front of the profile and steers the generation of the next vortex. In the case of a poor profile (depth $\approx$ thickness), the feedback is strong and the place where the next vortex starts is well defined. As a result we have a very stable vortex frequency. In the case of a good profile (depth $>$ thickness), the feedback is not so strong (larger distance between front end and rear end), the place where the vortices start is not so stable, and a broader frequency band is radiated."

This explanation sounds logical as long as slow travelling speed of the profile is considered. But when it approaches sonic speed, any acoustic upstream feedback from the trailing edge to the leading edge is bound to fail. This, according to the explanation mentioned, would mean that the profile noise decreases with the approach of sonic speed. The opposite, a rapid growth of the profile noise with the approach of sonic speed, has always been measured.

EXPERIMENTAL PROOF OF THE INDEPENDENCE OF THE PROFILE NOISE FROM THE TRAILING-EDGE VORTICES

The boundary layer creates the vortices leaving the profile at the trailing edge. If there is no boundary layer at the trailing edge, no vortices leave the profile. Thus, boundary-layer removal should greatly affect the profile noise in intensity and frequency if the explanation above is to hold true.

A single-blade propeller of 3 ft radius was spun in a sound-room. The blade consisted of a symmetrical profile of 12 per cent thickness, 1 in. profile depth at the tip, and 4 in. profile depth at the root. Provision was made on the outer 15 per cent radius of the blade to suck off the boundary layer near the trailing edge of the profile. The blade was spun at zero angle of attack. The flat response microphone of a sound-level meter was mounted concentrically with the blade at a distance of 6 ft from the blade. A pump could apply either suction or pressure to the duct inside the blade, which, in turn, was connected with 200 holes of 0·04 in. diameter spread out over the outer end of the trailing edge.

The profile noise of this blade was measured in the Mach Number range from 0·1 to 0·4, with the holes sealed as well as open, and the suction or pressure applied. The result was that neither suction nor pressure affected the profile noise previously measured with the holes sealed.

This measurement, in the opinion of the author, made it clear that the vortices at the trailing edge are not the primary cause of the profile noise.

TURBULENCE AHEAD OF THE PROFILE AS A POSSIBLE CAUSE

Noise pressure suddenly rises with the approach of sonic speed in a manner similar to that of the lift of a wing. On a short solid steel blade that is not affected by wing flutter and is driven extremely smoothly, one possible cause of angle-of-attack oscillation is the air turbulence ahead of the profile. With this idea in mind, calculations were made and tests carried out to determine the relation of ambient turbulence to profile noise.

One of these tests was to establish whether the concept of the turbulence, ahead of the profile, as a noise source, would explain the sudden rise in noise pressure with the approach of sonic speed. Fig. 1 shows the result of the peak noise pressure measured as a function of the tip Mach Number between Mach 0·2 and 0·8. Fig. 1 also shows two dotted curves, called "laminar" and "turbulent" respectively. The explanation of how these theoretical curves were arrived at is as follows:—

The average velocity vector of the turbulence $\Delta\mathbf{v}$ created by the previous revolutions must be a function of the drag coefficient of the profile C_D and the spinning speed of the blade v, or $\Delta\mathbf{v}$ is proportional to $C_D\, v$. This turbulence vector $\Delta\mathbf{v}$ will create oscillations of the angle of attack of the profile with the average amplitude

$$\Delta\overline{\alpha} = \Delta\mathbf{v}/v$$

or $\Delta\overline{\alpha}$ is proportional to C_D. These oscillations of the angle of attack will induce oscillations of the profile lift, or an acoustic pressure P_{ac}, which is proportional

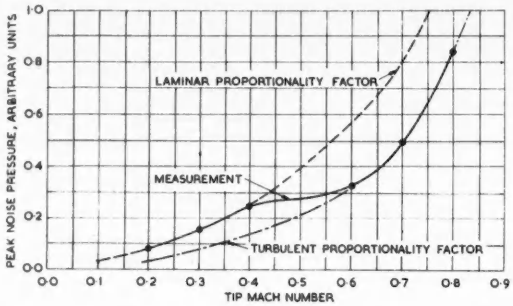

Fig.1. **Peak noise pressure of thrustless single-blade propeller *vs.* Mach number.**

to $C_D \times (d\,C_L/d\alpha)\,v^2$. In the laminar range of flow, C_D is proportional to $v^{-1/2}$. In the turbulent range of flow slightly above the transition, C_D = constant. The lift slope $(dC_L/d\alpha)$ of the profile in the subsonic range is a function of Mach Number. $dC_L/d\alpha$ is proportional to $1/\sqrt{1-M^2}$. Replacing v by M, we arrive at two proportionality factors of the noise pressure in the subsonic range: $P_{ac\ lam}$ is proportional to $M^{1\cdot5}/\sqrt{1-M^2}$ and $P_{ac\ turb}$ is proportional to $M^2/\sqrt{1-M^2}$.

The dotted curves in Fig. 1 represent these two theoretical expressions. It can be seen that the dotted curves well match the corresponding ranges of the measurement. The transition from laminar to turbulent occurs at a tip Reynolds Number of 3×10^5, which is in good agreement with aerodynamic profile and propeller experience.

The results derived from sound-room tests made it seem probable that the turbulence ahead of the profile was causing the profile noise. This would mean that the profile noise in flight cannot be studied in a sound-room, since the turbulence in a sound-room and the turbulence encountered in flight are not necessarily equal.

Profile Noise Measurement in Flight

Noise measurements on model propellers,[2] convey the impression that the rotational noise is predominant and that the profile noise is weak when compared with the rotational noise. This cannot be true under actual flight conditions for the following reasons: (1) Model propellers are always acoustically studied under static conditions, whereby the inner sections of the blades operate at reduced air speed, thus adding less to the total profile noise. (2) The much larger dimensions of full-size propellers shift the profile noise frequencies from the ultrasonic range to the audio range, while the rotational noise frequencies, which on models cover the audio range, shift towards the lower limit of the audio range. Thus, the profile noise under actual flight conditions may have much more bearing on the noise problem than is commonly realized. In order to clarify this situation, the following flight measurement was carried out:

The flat response microphone of a sound-level meter was mounted on top of a blimp. The installation of the microphone was acoustically checked for possible interference by the hull of the blimp. Since no such interference could be found, the blimp went up to an altitude of 5,000 feet, cut its engines, and slowed down to approximately 15 mph. An aircraft passed overhead at full throttle, equal to 340 mph, and at various relative altitudes of 300, 600, 1,200, 2,400, and 4,800 ft. At each relative altitude, the runs were repeated four to five times. The noise picked up during the passage of the aircraft was recorded and, later, frequency-analysed in the laboratory with the aid of a panoramic frequency analyser and a camera taking samples of the spectrum.

The results of the frequency analysis were surprising. The noise covered almost evenly the range from 200 to 3,000 cps at 300 ft range and 200 to 1,000 cps at 4,800 ft range. At no range did the spectrum show the sharp peaks and valleys that are characteristic of rotational noise. Thus, it was concluded that, under the specific conditions of this measurement, the profile noise must have been predominant. The centre frequency measured at short range was approximately 780 cps.

The centre frequency measured in flight made possible another check on the concept derived from the sound-room tests. In flight, the wake of the profile does not affect the turbulence ahead of the profile. Thus, in flight, the most reasonable assumption is that the atmospheric turbulence evenly covers all sizes of vortices. When assuming equal circumferential speed of all sizes of vortices, apparently the vortex with a diameter equal to the profile depth D adds most to the circulation about the profile and, therefore, creates the peak noise pressure. Since this vortex travels a distance equal to $2D$ from touching the leading edge to leaving the trailing edge, and creates 1·5 cycles of lift oscillation during this travel, the centre frequency should be approximately

$$Fr\ (\text{centre}) = \frac{1{\cdot}5\,v_0}{2\,D} = 0{\cdot}75\,\frac{v_0}{D}$$

where v_0 is the air speed of the profile and D is the profile depth.

When applying this formula to the blimp—aircraft measurement, the centre frequency measured equals that determined by the formula when the air speed and the profile depth at 80 per cent of the propeller radius are used. This is as close as an approximate formula and a single, though precise, measurement can come. The result again seems to support the idea that the profile noise is controlled by the air turbulence ahead of the profile, so that a fast-moving profile is an effective converter of weak ambient turbulence into powerful noise.

Conclusions

(1) The total noise of a propeller is composed of the rotational noise and the profile or vortex noise. The rotational noise has been theoretically studied. Measurements and theory are in good agreement. A current explanation[1] of the process causing the profile noise assumes the vortices leaving the profile to generate the noise.

(2) A sound-room test with boundary-layer removal on a spinning blade proves that the vortices leaving the profile do not primarily cause the profile noise.

(3) Additional sound-room tests seem to indicate that the ambient air turbulence ahead of the profile is primarily causing the profile noise.

(4) A flight test indicates that under actual flight conditions the profile noise may be much more important than model tests suggest. The fact that, based on the turbulence concept, the centre frequency of the profile noise measured in flight can be calculated, again supports the turbulence concept.

(5) If the new concept of the process causing the profile noise holds true, sound-room measurements of propeller noise cannot be applied to flight conditions. Also, full-size propellers must be much noisier when travelling in bumpy air than when travelling in almost turbulence-free air.

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Recent Advances in Soviet Machine Tools

By J. MANNIN.

In this concluding article of the series dealing with recent progress in the Russian machine tool industry, several new machines for the mechanical lapping of flat and cylindrical gauge surfaces are described. This is followed by a brief summary of investigations on general problems in machine tool design and development, recently published in the Russian technical press. These studies include the weight criteria for machine tools, as well as their stiffness characteristics. Criteria for the choice of speed changing control organs are mentioned. Several new machine tools are reported in various fields of metal cutting. The present position in the design of machine tool transmissions is briefly reviewed. Some new tools and machining methods are referred to.

MECHANICAL LAPPING

In a section of this series, dealing with measuring instruments and gauges (ENGINEERS' DIGEST, April 1953), mechanical lapping was mentioned as one of the main features of the current phase in the gauge making industry. Although lapping machines are a relatively new type of machine tool, Russian industry was compelled to urge the development of mechanical lapping, owing to the scarcity of skilled labour during the period of " hothouse " development of engineering production.

Scarcity of skilled hands imposes an increased use of inspection gauges and is therefore reflected in gauge output. Scarcity of skill in Russia is not confined to the fully apprenticed toolmaker, but embraces all grades of partial skill. The lapping process is usually performed by semi-skilled operatives, who require some months, if not years, of training which cannot easily be spared. We may therefore expect to find in Russia mechanised lapping processes which would not be justified on the ground of productivity alone. For this reason, the development of mechanised lapping elsewhere has been more cautious.

Another reason for the customary hesitation in introducing mechanical lapping is some confusion about the nature of the lapping process. It is easy to confuse lapping with some of the other finishing processes, such as grinding, polishing, honing, and superfinishing. Lapping is the oldest of these and consists in the removal of metal for the purpose of dimensional accuracy *and* surface smoothness. The cutting edges are those of a fine abrasive powder suspended in a liquid (the lapping compound), introduced between the workpiece and the lapping tool. This tool is responsible for the geometry of the lapped surface and the improvement of its dimensional accuracy. Usually, lapping tools are made of cast iron and their operative surface is flat, but an important branch of lapping is that of gearwheels, when the lapping tool is itself a gearwheel. Sometimes, lapping tools are made of abrasive material but even then the essential part of the metal removal is performed by the loose abrasive.

An essential feature of the lapping process is the removal of marks from previous machining operations. For this purpose, hand-lapping prescribes irregular movements and it is often believed that this irregularity, destroyed by mechanisation, is the very essence of the lapping process. This view is not entirely borne out by experience and it is found that a complicated regular movement, which in a short time interval appears irregular, can give results as good as, if not better than, hand lapping.

An original Soviet design for lapping of the flat surfaces of block gauges is shown in a diagrammatic perspective drawing, Fig. 1. The machine performs simultaneously the lapping of the top and bottom surfaces of a large number of block gauges. The laps are cast iron plates. The lower lap is fixed on the work table of the machine, which can be raised or lowered by a screw mechanism, operated by a handwheel. The upper lap is universally hinged and therefore self-orientating. The gauges are placed between the two plates and are guided in their motion by a set of steel tapes or strips with punched slots into which the gauges fit loosely. The upper lapping plate is loaded by another screw mechanism operated by a hand wheel. Its attachment to the screw contains an elastic element with a low-rate spring to maintain an even pressure, which is measured hydraulically and read on a pressure gauge.

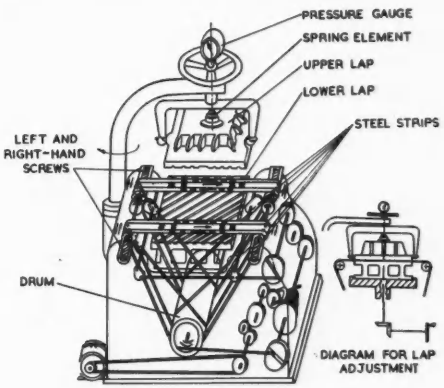


Fig. 1

The steel tapes are made to perform reciprocating motions in two directions. They are friction-driven by a drum, which is made to perform rotational oscillations through a crank mechanism, the throw of the driving crank being adjustable. This causes the longitudinal reciprocating motion of the steel strips. Their transverse reciprocation is imposed by flanged idling pulleys enveloped by the strips and caused to move axially by four screw mechanisms. These receive their rotational oscillating motion through another crank mechanism with an adjustable throw. It is seen in the illustration that the machine contains two sets of steel strips driven by the same drum, so that they perform their longitudinal motion together. By the use of left-hand and right-hand screws for displacing the idling pulleys axially, the transverse motions of the two sets of steel strips are in opposition to each other.

Both crank mechanisms for the longitudinal and the transverse motions of the steel strips are driven by a gear transmission from a 2-kW motor. The transmission contains a speed change gear-box. When the steel strips have performed a pre-set number of strokes, the whole machine is stopped by a switch triggered by a counter. The trajectory of a point on the surface of the workpiece in relation to the lapping plate is a straight line at a small angle to the direction of the longitudinal motion. Before starting the process, the lapping plates are charged with the abrasive compound. For this operation, both plates have to be removed from the machine. Charging must be repeated after every two or three batches of gauges.

These machines have been in service in the " Kaliber " Plant for about two years and, it is stated, have completely eliminated hand lapping of block gauges. The usual extravagant claim is made that productivity has been increased several dozen times, a claim which,

as explained on several occasions in these articles, is as spectacular as it is meaningless. An interesting detail in the application of these machines is the practice of disengaging the transverse motion towards the end of the lapping process. It is claimed that this practice creates longitudinal lapping marks and thereby ensures better adhesion of the gauge faces. This claim is in marked contrast with the accepted conception of lapping and the belief that the adhesion of block gauge faces is due, in part at least, to molecular forces.

The most important group of lapping machines is characterised by a rotational main lapping motion with superimposed additional motions of various kinds. Such machines naturally have round lapping discs, usually mounted on vertical axes. The work is loosely held in a workholder or separator disc. The simplest type in this group is the "Hock" machine with stationary lapping discs and a separator describing a cycloidal motion, driven by mechanism which consists of several planetary pinions pivoted on rotating cranks and engaging the toothed crown of the separator disc. Another type with rotating lapping discs and a separator consisting of a complete epicyclic gearing, of which the planet wheels carry the workpieces, is used in Russia, as elsewhere, for lapping the faces of piston rings.

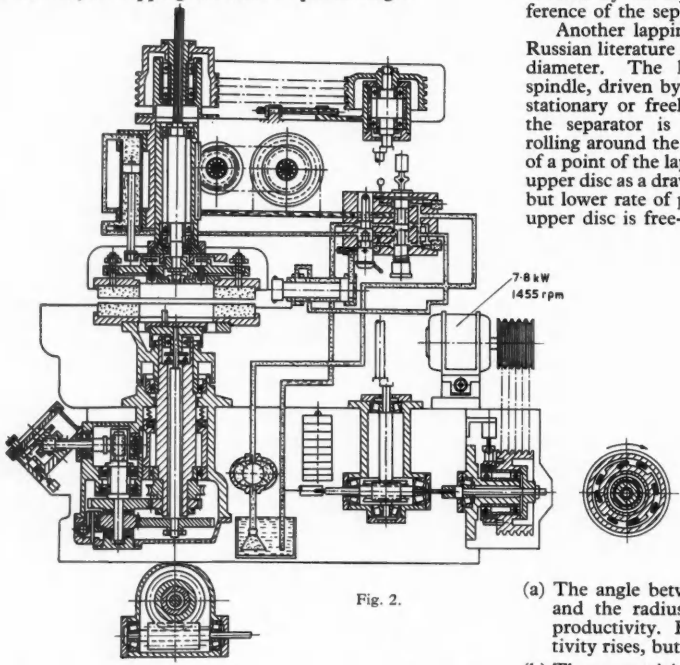


Fig. 2.

A more elaborate lapping machine of original Russian design is shown in semi-diagrammatic sectional form in Fig. 2. It is intended for the lapping of both flat and cylindrical surfaces. The laps are discs of 600 mm diameter mounted on two vertical spindles. Each disc is rotated by a drive starting from a 7·8 kW motor through V-belt transmissions, universally coupled shafts and worm drives. The upper lapping plate is universally hinged to its spindle, which can be moved axially by two hydraulic jacks symmetrically disposed about the spindle. The lapping disc is advanced towards the work and pressed against it by the hydraulic jacks. The spindle is balanced by a counterweight.

In this machine, the lapping discs are made of a bonded abrasive substance and the machine contains a trueing mechanism, which is hydraulically actuated.

The separator disc is driven by an eccentric pin mounted on a plate inside the central hole of the ring-shaped lapping disc. The plate receives a separate drive by means of a concentric spindle inside the hollow axle of the lower lapping disc. When lapping flat surfaces, the separator is driven in the same direction as the lower lapping disc but at a different speed. In lapping cylindrical surfaces, the eccentric pin is stationary and merely constitutes the pivot for a free rotation of the separator. The speed of this rotation is determined by the difference in the rotational speeds of the lower and the upper lapping discs (about 8 per cent). Usually, the cylindrical workpieces are placed in the separator with their axes at an angle to the separator radius. This causes a more rapid removal of metal. The angle chosen is between 5 and 30 degrees. In following the kinematics of motion in this arrangement, it will be found that the ends of the cylindrical workpieces have larger components of sliding motion than their centres, and the tendency exists for generating a cambered cylindrical surface (" barreling "). With long cylinders, it is necessary to counteract this tendency by arranging a large enough eccentricity of the separator pin to allow the workpieces to emerge partly beyond the area covered by the lapping discs over part of the circumference of the separator.

Another lapping machine of this type described in Russian literature has cast iron lapping discs of 400 mm diameter. The lower disc rotates with the lower spindle, driven by the motor. The upper disc is either stationary or freely rotating. In lapping flat surfaces, the separator is driven by three planetary pinions rolling around the stationary sun wheel. The trajectory of a point of the lapped surface appears on the stationary upper disc as a drawn-out epicycloid. A higher accuracy, but lower rate of production, can be achieved when the upper disc is free-wheeling.

A very simple lapping machine has been found satisfactory for the lapping of taper rollers. It has a lower stationary cast iron disc mounted on the bed of the machine in a spherical cup for free orientation. The upper lapping disc is of bonded abrasive substance and rigidly mounted on the upper spindle driven by an electric motor. The rollers are placed in a separator, freely rotating about an eccentric pin.

Experiments were carried out with the machine illustrated in Fig. 2. Two of the conclusions from these experiments may be of interest :

(a) The angle between the axis of the lap of the roller and the radius of the separator has an effect on productivity. Between 10 deg. and 30 deg. productivity rises, but after this falls sharply.

(b) The eccentricity of the separator increases productivity, provided the components do not leave the surface of the lapping disc. When this is necessary for accuracy in long components, a sacrifice must be made in the rate of metal removal.

Another subject of experiment was the composition of the abrasive liquid. The smoothness of the surface depends almost exclusively on the particle size of the abrasive powder. The effect of additions of various substances was studied singly and in combination. Many compositions were tried and, of these, the two best are :

(1) Paraffin, 20 litres ; stearine, 0·5 kg ; abrasive powder, 0·95 kg.

(2) Petroleum jelly, 20 litres ; stearine, 0·7 kg ; abrasive powder, 1·05 kg.

The experiments also included the effects of lapping on the geometric accuracy, surface roughness, and micro-hardness. In cylindrical lapped components, the ovality is reduced and the roundness is improved. Surface roughness values between 0·06 and 0·27 micron r.m.s. are obtained, depending on the grit of the abrasive powder. After a removal of metal by lapping to the extent of from 7 to 10 microns, the layer of increased micro-hardness, formed as a result of local quenching in the grinding process, is removed.

General Studies

The central planning of machine tool development programmes encourages attempts to establish criteria for quality applicable to the large variety of designs. Among the currently prominent problems, much emphasis is placed on a reduction in weight. The benefits of weight reduction in machine tools, provided other qualities are maintained, are manifold. Some are of special importance in the Soviet Union. In general, apart from the saving in scarce materials, appreciable reduction in cost is obtainable, since the expenditure on raw material constitutes a considerable proportion of the total cost of many machine tools. The reduction of mass in moving parts reduces the dynamic loads or permits the increase of allowable accelerations, thereby leading to increased productivity. In Russia, most machine tools are manufactured in a few central plants and transported across vast distances. In the industrialisation of some outlying regions, transport costs constitute a large proportion of the investment in the total plant.

Clearly, establishing weight criteria for machine tools is a much more difficult problem than in other fields of engineering design. Machine tool design is characterised by continual improvements and increasing demands for accuracy and standards of productivity. Machine tools, which are made in a variety of types unknown in any other branch of mechanical construction, cannot, therefore, be judged by a single measure, such as the horse-power to weight ratio in prime movers.

Nevertheless, for a given geometry of the cutting tool and a given cross-section of the cut, the volume of the metal removed, which is a measure of productivity, is proportional to the power consumed, and roughly proportional to the power imput. Thus, the weight per unit of power of the main drive is a significant characteristic quantity. It varies between 0·2 and 4 tons/kW. Some typical values for engine lathes are 0·32 tons/kW for a 1934 Soviet design and 0·24 for a 1949 model. Horizontal milling machines of Soviet origin have developed from 0·34 to 0·19; cylindrical grinding machines from 0·38 to 0·35.

One of the fundamental inconsistencies of the weight/power criterion is the fact that its technical justification is based entirely on productivity during the machining period. The essence of many modern improvements is the reduction of non-productive periods. The weight/power criterion ignores these essential properties of the machine and its degree of labour saving in permitting operation without attention. For this reason, a second specific weight criterion is proposed, which is a refinement of the first. The weight of the machine is referred to the effective power used in cutting, and to two efficiency factors, of which the first is defined as the ratio of the machining time to the total time per piece and the second as the proportion of the total time during which the machine can be left unattended.

Another weight criterion is physically based on the effective use of material in withstanding the essential loads which the machine has to carry in performing its task. Thus, the weight of the load-carrying parts of a drilling machine can be referred to the axial pressure of the drill. Typical values are between 0·5 and 1·1. The weight of the main transmission of a lathe can be referred to the torque of the work spindle. Typical values are between 3 and 8 kg/kgm. In this connection, it may be interesting to note that the specific weight of Russian gear-boxes is in the region of 1·8 kg/kgm for lorries, and 2·5 for passenger cars.

An additional weight criterion is necessary to judge the weight reduction brought about by a complete change of basic lay-out. Such a change would not necessarily find its expression in the previously considered weight criteria. Another criterion is therefore proposed, which expresses the ratio of the weight of all those components of the machine which are directly concerned with the process of metal cutting, to the total weight. This criterion is the foundation for some of the successful efforts in weight reduction involving departures from orthodox principles, such as the replacement of the moving work table by the moving grinding wheel spindle in heavy cylindrical grinding machines, and the creation of portable machine tools, the centreless shaft peeling machine, and the centreless grinder.

Apart from the evaluation of weight criteria, investigations of stiffness and temperature effects have been published recently in Russia. It is not easy to establish a universal stiffness criterion or to find a method for its measurement. An interesting approach is the suggestion to measure the stiffness of a horizontal milling machine by its vibrational frequencies. Methods of design for increasing stiffness at a low expenditure in weight have received great prominence in Soviet technical literature, but do not reveal ideas unfamiliar elsewhere. Investigations of heat deformations also contain few innovations. However, the following points are perhaps worth mentioning: The delivery of the coolant pump in litres per minute should be 5-10 times the power in kW of the main drive; an increase in the cutting speed causes a larger proportion of the heat developed in cutting to be carried away in the chips; abrasive machining should be carried out with as large a wheel as possible.

A general analysis of gear change mechanisms contains an attempt to establish criteria for the evaluation of such gears in terms of their effect on productivity. These criteria narrow down the choice between different lay-outs and sometimes even single out one solution from all others. Gear-changing mechanisms can be judged by the degree of centralisation, considered perfect when a single control organ is used; by the selectivity factor, which measures the average degree of approximation to the desired speed; by the duration of the average gear changing operation; and by a measure of the number of the actual mechanical engagements in the mechanism, thereby determining the resulting wear. It is recommended that in each specific case these criteria should be given their appropriate weight and that subsequently all alternatives of gear box layouts should be evaluated with the help of these factors.

Of fundamental interest is an investigation of all the methods suitable for forming bodies of revolution. It is claimed that 45 different schemes are conceivable, of which 28 have a present or potential practical importance, and have been examined experimentally to establish formulae for the relations between the main parameters of the cutting process.

Random Details

Apart from the new machines described in some detail in previous articles of this series, several other machines, having some new features of technical interest, have appeared in Soviet factories. Among standard machine tools, an important creation is a centreless grinder with an exceptionally wide grinding wheel (800 mm). It is claimed that this development can increase productivity 3½ times, compared with centreless grinders of standard wheel width. This centreless grinder has been found suitable not only for

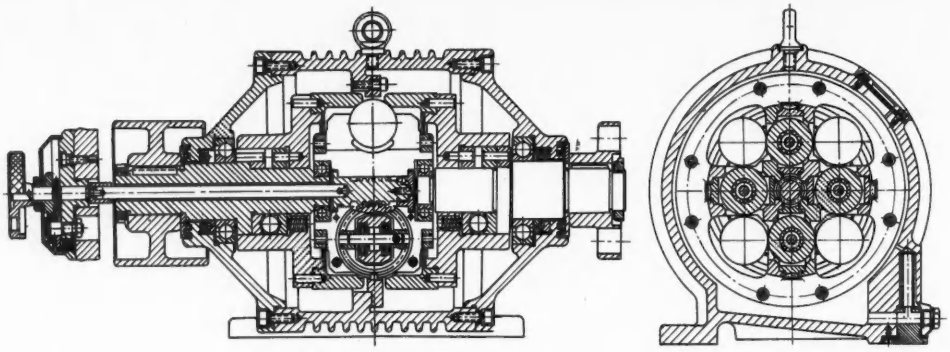


Fig. 3

the grinding of long shafts, but also for the simultaneous grinding of a number of short components and in this role has been incorporated with success into automatic production lines for the manufacture of pistons and ball races.

Machine tool designers in Russia, as elsewhere, have continued to devote attention to the development of improved transmissions. The ideal transmission should cover the whole range of powers occurring in machine tools, should offer infinite variation of speed within a large range, should have a high efficiency, should be robust and durable, and should be inexpensive. Practical transmissions do not fulfil all these demands and better compromises are continually sought. From the technical point of view, the electro-mechanical transmission, employing the cross-field generator (amplidyne) is the most satisfactory. It has been embodied in a single spindle automatic lathe brought out in 1952, where the work spindle transmission is under fully automatic control. However, this transmission is recognised as a luxury, too expensive for any but special purposes.

Another infinitely variable electrical transmission, with thyratron-controlled d.c. motors, appeared last year for a power of 4·5 kW under the " trade name " Elir. Again it is considered that the cost of such a transmission is not justified and Russian industry has not yet produced thyratron valves of sufficient power for such a drive.

The use of hydraulics in Russian machine tools was the subject of a general survey of the present status and future problems of this transmission medium. These follow closely the general lines of European and American machine tools in the design of pumps and their control gear, of distribution panels, reversing gear and hydraulic copying devices (discussed earlier in this series). The problem of creating a rotary hydraulic drive for the main transmission is still regarded as unsolved. Several minor applications appeared many years ago, such as in the rotation of the boring bar of a diamond tool boring mill, the rotation of the work spindle in a centreless grinder, the rotation of the lead screw in a universal thread grinder, and the rotation of the work table in a surface grinder. However, hydraulic transmissions for powers of several kW are not yet considered a practical proposition on account of their low efficiency, lack of durability, and high cost. Nevertheless, development work on these units is still being carried out, and a 15-kW unit is under test by ENIMS (Central Machine Tool Research Institute).

In the mechanical friction-type infinitely variable speed gear, Russian constructors have made some notable advances. A unit transmitting up to 2·8 kW with a range of speed variation of 1 in 12 is shown in Fig. 3. It is an original design created by ENIMS and is said to have a satisfactory efficiency and to require little attention in operation. It will be seen that the gear contains two cups, each coupled to the input and output shaft respectively, and having inner conical friction faces, between which four balls maintain a friction drive. The balls are in contact at a certain point with rollers, which determine the axis of rotation of the balls. A change in the speed ratio is carried out by inclining the guiding rollers with the help of the worm drive control gear seen in the illustration.

A remarkable feed motion transmission was shown last year applied to all the feed motions of a universal boring mill. This electro-mechanical transmission of the amplidyne type with feed back ensures a speed control of 1 in 600 and has considerably facilitated the installation of semi-automatic control. Limit stops in this machine can arrest all feed and adjustment motions with an accuracy sufficient for all ordinary boring operations.

A high speed gear shaper of the " Fellows " type for straight and spiral spur gears of small size and pitch is now in full production. Several additions to the range of screw cutting lathes have been made. These include, in particular, a high precision lathe for the cutting of lead screws, some features of which were mentioned in the discussion of measuring instruments, and several new designs of high production screw cutting machines. Of particular importance for the oil industry is a machine for the cutting of tapered pipe threads. These are special machines designed to overcome the handicap of long clamping times, which are of particular importance in this work, owing to the short duration of the cutting itself. The machine is said to have a clamping time of 5 to 6 seconds.

Some gaps in machine tools for very large components have been filled by grinding machines for steel rolling plant rollers up to 1·5 m in diameter and up to 6 m in length. A heavy gear hobbing machine for gears up to 5 m diameter is now available. It is said to be capable of heavy cuts, up to 12 mm advance per revolution of the work, and the design of the transmission for the generating motion is such that the transmission is said to be completely unloaded of the cutting effort.

Several new types of cutting tools and machining methods have been introduced or tried out during the period covered by this review of recent progress. Some of these ideas have been more fully reported elsewhere in these pages, e.g. tools tipped with ceramic materials (ENGINEERS' DIGEST, February 1953) and the use of a plastic mass in the clamping mechanism of fixtures (ENGINEERS' DIGEST, April 1952, January and April, 1953).

Among other new tools of interest, are twist drills, manufactured by the twisting of a rolled high-speed steel profile ; these are stated to save 35 per cent of material and to be produced at 1/30th of the labour cost of twist drills with milled flutes.

Apparatus for Measuring the Shrinkage of Plastics during Polymerisation

By G. DIXMIER. (From *La Recherche Aéronautique*, No. 31, January-February, 1953, pp. 49-53, 6 illustrations.)

The new composite plastics for moulding of components at ambient temperatures have the serious disadvantage of considerable shrinkage during polymerisation. A test apparatus and procedure have been devised to determine, inter alia, *the amount of shrinkage to be allowed for in the design of moulds and the conditions for minimum shrinkage with different types of fillers.*

THE manufacture of semi-stressed components from composite plastic materials by moulding at normal workshop temperature has reached the stage of prototype production. There are several advantages obtained by the use of such plastics, but some difficulty has been experienced in manufacture, owing to the fact that the resins used as cements tend to contract considerably while passing from the liquid to the solid state. This shrinkage (which may be considerable) is due partly to polymerisation and partly to the high thermal expansion coefficients of resins.

Dimensional accuracy of moulded parts can be obtained only if the shrinkage effects are fully under control. Furthermore, in connection with the mechanical properties of composites, knowledge of the factors governing the changes in volume is of importance, as volumetric changes occurring during the setting result in internal stresses, which may produce failures of the components, while, in other instances, microscopic or sub-microscopic cracks may develop.

It might be suggested that volume variation could be assessed from the cubic expansion coefficients of polymers. However, the use of such coefficients taken from general literature tends to cause misleading results. Spencer and Boyer,[1] Kolb and Fizard,[2] and Kovacs,[3] have shown that it is useless to indicate figures of mean cubic expansion without adding details of the thermal history of the specimen and its period of exposure to each temperature. Kolb and Fizard, in particular, have indicated that the dilatometry at different temperatures of resins in the solid state should be related to the crystallisation of the polyester, i.e., to the structural evolution in the solid state. These remarks show that investigations must be carried out thoroughly, without preconceived ideas regarding the separation of the phenomena of structural evolution from those due to normal expansion.

The object, therefore, has been to develop an apparatus permitting measurements to be made of volume variations of resin masses in the solid state, particularly also during the transition stage from a paste to the solid state, i.e., during the setting. It will be shown in the following that the equipment and procedure which have been developed can also be employed for the study of the kinetic processes in polymerisation.

DESCRIPTION OF APPARATUS

Basically, the apparatus (see Fig. 1) is a mercury type dilatometer. The special feature is that the monomer substance is placed in a flexible rubber bag impervious to liquids, situated in the apparatus and immersed in the mercury.

By means of this arrangement, the resin is not restricted in its volume variations. It is easy to verify that, if a sample of monomer is contained in a solid-walled vessel, it will adhere to some extent to the walls during its hardening. This adhesion causes internal stresses, which may produce cracks, into which the dilatometric liquid cannot penetrate. Under such conditions, it is possible to measure only the apparent shrinkage. Moreover, owing to the internal stresses, the simultaneous occurrence of the chemical process and shrinkage is modified, and this may alter the shrinkage curve as a function of time.

The bag is made of pure rubber without sulphur and is placed in a fixed position in such a way that all parts of its surface will be subjected during the tests to the hydrostatic pressure of mercury in substantially the same manner.

This pressure is of the order of 4 to 500 grams/cm^2 and has no effect on the polymerisation reaction or on the volume measurement, since resin, whether liquid or solid, is practically incompressible. The pressure is applied to ensure that the elastic membrane will follow all the changes in volume of the resin which it contains. These changes are measured from the displacement of a column of mercury in a calibrated tube.

The apparatus is immersed in a thermostatic tank filled with water and containing a heating coil, a contact thermometer and a stirrer. The mercury-type volume measuring apparatus comprises three glass components, viz., a cylindrical body, a filling burette and a stopper, as shown in Fig. 1.

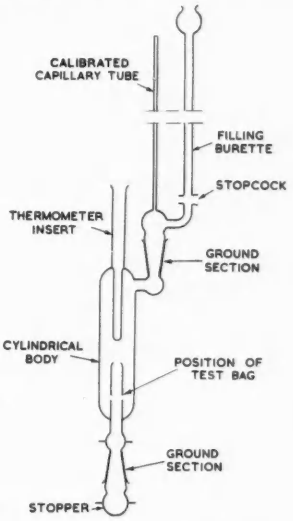


Fig. 1. **Layout of volume measuring apparatus.**

The cylindrical body contains the mass of mercury and the rubber test bag. In the upper part of the body an L-shaped tapered tube section is arranged on one side to take the burette used for filling. Also, in the upper portion and in the axis of the cylinder is a glass insert for the thermometer, which indicates the temperature in the vicinity of the resin during the reaction. The lower part of the cylindrical body contains an inner tube with radial holes, which receives the rubber test bag. This tube has an extension, ground to take a stopper.

TEST METHOD

The empty apparatus is dismantled and its components are placed in an oven at the required thermostat temperature. The empty test bag, the mercury and the previously weighed resin (about 1 gram) are also heated to avoid errors due to volume variations of substances with dissimilar temperatures.

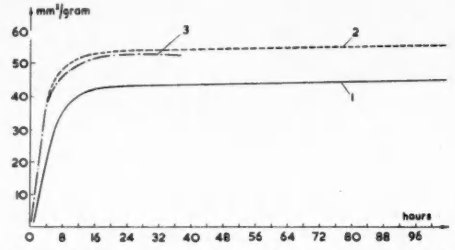


Fig. 2. **Shrinkage curves of a Laminac type resin, at 30° C.**
(1) 1% Catalyst. (2) 1·5% Catalyst. (3) 2% Catalyst.

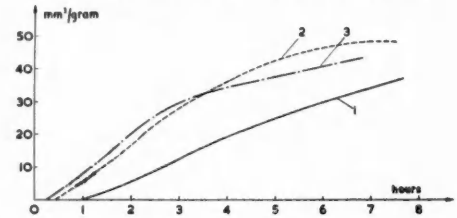


Fig. 3. **Initial portion of shrinkage of a Laminac type resin, at 30° C.**
(1) 1% catalyst. (2) 1·5% catalyst. (3) 2% catalyst.

Before being heated, the rubber bag is immersed in a solution of polyvinyl alcohol which, after drying, leaves an insulating film on the rubber. This avoids subsequent contact between the resin and the rubber constituents, with the possibility of affecting the hardening process of the resin. The hardening agent is added to the resin, which is placed in the rubber bag. The bag is then closed to avoid air bubbles in the resin or between the resin and the bag. Closure is effected either by means of a cork impregnated with polyvinyl alcohol or simply by tying the neck of the bag with thread or an elastic band. After the filling operation, the bag is inserted and the lower part of the apparatus is closed by means of the stopper. The filling burette is mounted at the upper end of the apparatus, which is then immersed in the thermostatic tank. The burette is filled with mercury, which passes into the cylindrical body, while vacuum is created through the calibrated tube.

By careful operation, the apparatus can be filled up to a mark on the graduated tube without leaving any air bubbles in the mercury. The assembling and filling operations take about 15 minutes. Some uncertainty remains regarding the beginning of the reaction, since measurements are taken of the volume variations indicated by changes in level of the mercury column in the calibrated tube, and these can be observed only some 15 minutes after insertion of the hardening agent and the catalyst.

Measurement of Shrinkage

Readings of the mercury level are taken at 15 min intervals, and later at hourly intervals, until the level remains constant.

If A is the mean cross-sectional area [mm²] of the capillary, and Δh the change in level (mm), then $A\Delta h = \Delta V$ is the total volume contraction [mm³]. The relative shrinkage R is determined from

$$R = 100 \times \gamma \, \Delta V / P \text{ per cent},$$

where P is the weight of the resin [grams] and γ its specific weight [gr/mm³].

Graphical Representation of Shrinkage

Two different types of diagrams have been used to represent shrinkage as a function of time :

Diagrams of the first type: The volume variation per gramme of resin is plotted against time. Typical examples are the curves of Figs. 2 and 3, obtained with a polyester resin of American origin of the Laminac type. Fig. 2 shows the total shrinkage over a period of 100 hours. The shrinking process is apparently completed, from the point of view of practical applications, but there is still a very slight slope which can be detected with more sensitive equipment. These curves clearly show a rapid change during three to five hours.

In the first period, the shrinkage reaches 85 to 90 per cent of its full value. This may be regarded, therefore, as the "setting time," after which the component can be removed from its mould. The second period represents the "hardening time," at the end of which the material has acquired the mechanical properties required in its use.

Fig 3 gives the same diagram but with a wider time scale, for a closer study of the polymerisation reaction. It clearly shows a preliminary period which is more pronounced when the proportion of catalyst is smaller. Referring again to the curves in Fig. 2, it is seen that the limiting value of the total shrinkage increases with the proportion of catalyst, i.e., as a function of the quantity of polymers formed in the reaction. In this instance, there seems to be an optimum proportion of catalyst beyond which there is no further increase in shrinkage and hence in the degree of polymerisation ; only the shrinkage rate is higher and this is in agreement with previous information on the kinetics of polymerisation. Furthermore, the existence of an optimum ratio of catalyst for this type of resin has also been verified in practical applications.

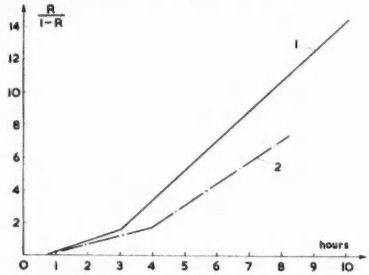


Fig. 4. **Diagram of $R/(1-R) = Kt$, for a Laminac type resin, polymerised at 30° C.**
(1) 2·0% catalyst (1st test). (2) 2·0% catalyst (2nd test).

Diagrams of the 2nd type: These are in accordance with a representation suggested by Adams and Powers in their study of the polymerisation of linseed oil, which also occurs by means of double bonds. If ΔV_t = volume contraction at the time t, and ΔV_{max} = maximum contraction, then the relative shrinkage is

$$R = \Delta V_t / \Delta V_{max} \qquad (0 < R < 1)$$

and, assuming the reaction is of the second order, the expression

$$R/(1-R) = K\,t$$

is obtained by integrating the differential equation $dR/dt = K\,(1-R)^2$, where K is a constant. Thus, $R/(1-R)$ can be plotted as a function of time.

In view of the complex nature of the polymerisation reaction, no assumptions can be made, but the method of representation is used because it indicates divergences between two tests which give apparently identical curves, when represented as shown in Fig. 2. Using the same type of Laminac resin with the addition of 2 per cent of catalyst, these tests give two pairs of straight lines which are clearly shown in Fig. 4. It was found that this scatter is due to different conditions in

(*Concluded on page* 232)

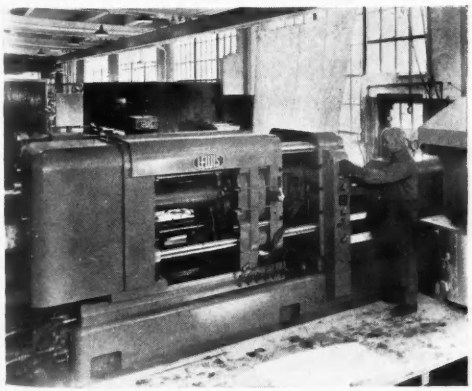
New Materials, Processes and Equipment

DIE-CASTING MACHINE

Lewis Welding & Engineering Corporation of Bedford, Ohio, U.S.A., have developed a new die-casting machine which, it is claimed, should produce bronze, aluminium or magnesium castings with a very low percentage of rejects. The machine employs the cold-chamber method of injection, but introduces a new principle of injection which provides for a rapid fill while the metal is molten, and a high final squeezing pressure. This prevents the occurrence of shrinkage voids as the metal freezes and of excessive porosity due to the trapping of air and gases in the dies.

The new Hydra-Lock clamping system which develops a final pressure of 650 tons for clamping the dies together, is incorporated. A small hydraulic cylinder advances the platen on which is mounted the moving half of the die, and another hydraulic cylinder, 28 in. dia., closes the die and supplies the locking pressure.

The machine develops an injection pressure of 100,000 lb which is accomplished by a special hydraulic two-stage injection system. Two pistons are employed, one being concentric with the other. The piston of small diameter travels at speeds variable up to 810 fpm to fill the die cavity with molten metal. When the die is filled, resistance is encountered and the piston of large diameter is brought into operation. A total pressure of 100,000 lb is developed by means of the 2100 psi available in the hydraulic system. This is the maximum pressure that can be exerted but it can readily be adjusted to suit casting requirements. The injection speed and pressure can be adjusted to conform to the actual requirements of the die cavity and the natural flow of the molten metal into the die cavity. Due to the short stroke of the hydraulic clamping cylinder and the small oil consumption of the injection system, the complete casting cycle is accomplished with only 4 gal of oil. This is accompanied by minimum loss of power and the elimination of the necessity for oil cooling devices.

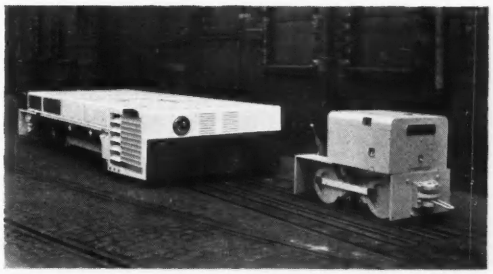


High factors of safety have been employed in the mechanical design of the machine and stresses are kept to a minimum to ensure accurate alignment. The machine bed, frame and platens are made from heavy rolled-steel plate to give maximum strength without excessive weight. The machine has been made as compact as possible within the limitations of ready accessibility of the working parts and its total weight is 29,000 lb. The hydraulic unit is furnished as a separate unit in order to make installation as convenient as possible for the user and to reduce the overall length of the machine to a minimum. The hydraulic unit may be set anywhere with reference to the machine itself, provided that the length of piping is not excessive.

NEW DIESEL LOCOMOTIVE FOR MINES

A revolutionary type of diesel locomotive has been introduced by the Hunslet Engine Co. Ltd., of Leeds, specially for underground service. It is only 39 in. high, but is able to start and pull well over 600 tons along the level at 5½ mph. It has a four-speed gearbox, giving track speeds of 5½, 7, 9½, and 14 mph, and is powered by a 170 hp horizontal oil engine. Maximum tractive effort is about 11,000 lb and the weight 37,000 lb. The 24-inch wheels are spread over a base of 7 ft, the overall length being 18 ft. Full flame-proofing and exhaust conditioning are included. The height is believed to be the lowest ever achieved with a locomotive carrying its own source of power. This particular model is stated to be the most powerful in the world.

The illustration shows the new locomotive, compared with a more orthodox model, fitted with a 15 hp air-cooled oil engine.



ELECTRONICALLY WELDED INDUSTRIAL CLOTHING

The remarkable resistance to chemicals afforded by P.V.C. plastic sheet materials, coupled with their toughness and light weight, makes them an obvious choice for industrial protective clothing. To obtain the full advantage of the material, however, it must be fabricated by means other than stitching. This requirement has now been achieved by Plysu Products Limited, of Bletchley, in their range of protective clothing, which is fabricated throughout by H.F. welding. This company was the first to undertake electronic welding of plastics on a commercial scale, and is believed to be the largest manufacturer using this system in Great Britain.

From elaborate quick-release suits for workers in radioactive atmospheres to simple boiler suits, aprons and gloves, the Plysu range of welded protective clothing covers most industrial needs.

TRIMMING PRESS

The development of a new line of trimming presses, especially designed for rigidity and quietness of operation, is announced by E. W. Bliss Company, Canton, Ohio.

The single-geared presses in this line are of four-piece frame-welded steel construction with twin herringbone driving gears running in oil and equipped with rim-type, oil tight gear guards.

Among the features of the new Bliss No. 210

Power take-offs and clutches

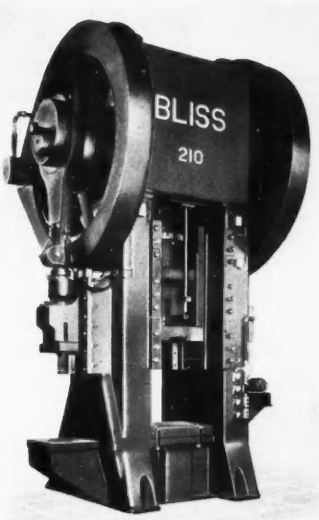
Rockford British-made heavy-duty clutches and power take-offs are being used to a rapidly increasing extent on industrial petrol, paraffin and diesel engines, and on machines such as portable air compressors, power shovels, pumps, excavators, winches and cranes.

These heavy-duty clutches have several important features, including an actuation system which maintains the clutch in either the engaged or the disengaged position, and in each case without running thrust.

The upper illustration shows the MacLaren power pack, embodying a Rockford unit with single-plate clutch.

Descriptive booklet, with list of standard sizes, on application from :—
BORG & BECK COMPANY LIMITED,
LEAMINGTON SPA, ENGLAND

ROCKFORD



Trimming Press illustrated are an air counterbalance concealed in the uprights, automatic lubrication and the Bliss fast-acting, cool-running friction-type clutch. Another feature of this press is the friction clamp slip-type knockout which prevents accidental breakage of dies or other parts, by permitting the friction clamp to slide on the knockout bar whenever the bar has not been adjusted to the proper height.

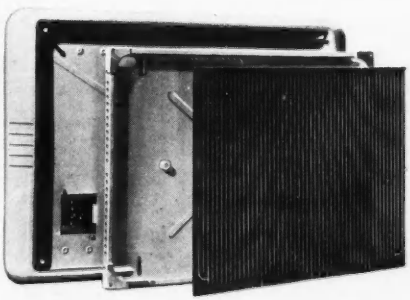
The press operates at a speed of 35 strokes per minute with ram capacity of 400 tons. The slide has a 16-inch stroke with a motorised adjustment of six inches, and the trimming attachment has a 10-inch stroke with a 4-inch adjustment. The press is 18 ft. high, has floor space requirements of 94 in. × 86 in. and weighs approximately 45 tons.

GLASS PANEL RADIANT HEATER

A radiant heating system originally developed in France, has been perfected by Glassheat of Canada Ltd., Toronto, and appears to be increasing in popularity in North America as an efficient method of heating.

The heating unit, shown in the illustration, consists of a 16 × 24 in. rectangular glass panel, $\frac{1}{4}$ in. in thickness, with an aluminium grid fused to the back of the glass. An aluminium reflector is placed between the glass panel and the back of the mounting frame. Electricity passing through the aluminium grid raises the temperature of the grid and glass panel to as high as 340° F. Temperature differentials cause air to circulate through vents at the top and bottom of the panel, thus heating by means of a combination of convection and radiation. The heated glass panel, when warm, acts as a radiating surface. Even though the surface temperature of the glass at times approximates 300° F, contact with the glass will not cause burns, owing to its insulating qualities. Thermostats provide automatic operation to assure uniform temperatures.

There are two basic types of panel: one designed for recess into the wall and one for installation on the surface of the wall in completed structures. A junction box at the rear of the mounting frame provides means of connecting the device to electric circuits. Each panel is rated at 1000 W, and the glass used is virtually unbreakable.



WIRE DRAWING DIE MEASUREMENT

The British Iron and Steel Research Association, London, announce the presentation of their BISRA Profiloscope for the measurement and inspection of wire drawing dies.

The Profiloscope, which is manufactured by Longworth Scientific Instrument Co. Ltd., of Abingdon, is a simple and robust die inspection tool, which makes possible rapid and accurate measurement of die angle to ± 0·5 deg., and qualitative examination of the sizing portion of the die and of the zone where the reduction taper blends into the parallel. It can also be used for exceptionally rapid quality control of die profiles.

The instrument is based on a very simple principle in which a narrowly divergent light beam passes through the die placed on a tilting table, on to a screen placed below. When the die is horizontal, a small bright circle of light is seen, outside which is a dark band, outside which again is a narrow band of light, all these being concentric one with another. The innermost circle is due to direct illumination through the die hole and the outer to internal reflection from the tapered region of the die. Should there be any changes in taper, these will immediately become evident as more concentric light circles, one for each change in taper, while a gradual change in taper causes a wider band of light. Any scoring will show as light radial lines radiating from the centre circle and the effect may be intensified by tilting the die. Inspection of the die interior is therefore very simple. Two examples of these effects are illustrated.

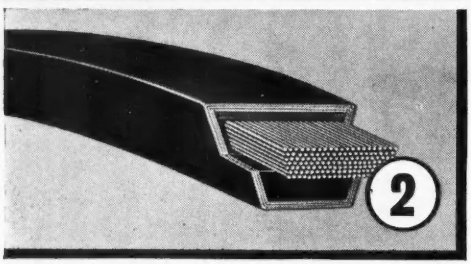
FOR HARD WORK AND PLENTY OF IT–

GOODYEAR BELTING AND HOSE

IN THE Goodyear range you will find conveyor belting for every type of service, transmission and V-Belts to suit every condition of loading, hose to meet every industrial need. Built with the accumulated knowledge of over 50 years' pioneer research and practical engineering, Goodyear Industrial Rubber Products give longer wear, more dependable service. And, above all, they keep operating costs to a minimum. That's why it pays to specify Goodyear.

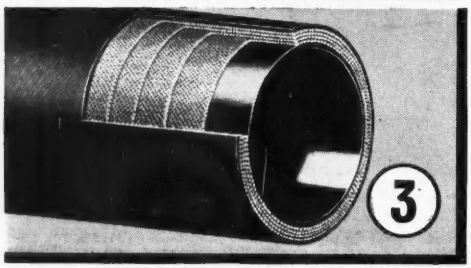
1 GOODYEAR CONVEYOR BELTS

The "Stacker" belt illustrated here is designed for installations which handle highly abrasive materials. Its tough resilient cover "gives" under impact and resists cutting. High grade bonding between the plies prevents separation under severe flexing. It is proofed against the ruining effects of mildew.



2 RAYON CORD V-BELTS

Goodyear Rayon V-Belts are ideal for all industrial drives, especially those longer, normal speed drives where shock loads occur. They stand up to sudden jolting loads that would break ordinary belts, for they contain strong, resilient, high-tensile rayon cords which "give" without snapping.



3 WRAPPED PLY HOSE

This Goodyear Hose is built from high-grade rubber tube wrapped in tough rubberized fabric for greater strength. A protective cover of bruise- and abrasion-resisting rubber assures lasting wear. And scientific arrangement of the fabric plies minimizes kinking. Intended for general service this wrapped ply hose gives long life under the most arduous conditions.

GOODYEAR

INDUSTRIAL RUBBER PRODUCTS

TRANSMISSION BELTING · V-BELTS · CONVEYOR BELTING · INDUSTRIAL HOSE

The Goodyear Tyre and Rubber Co. (Gt. Britain) Ltd., Wolverhampton

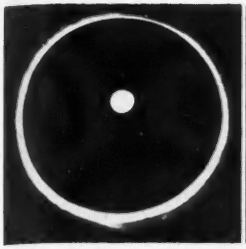
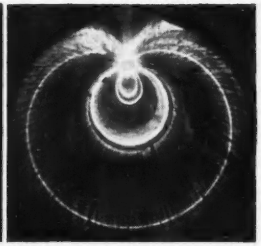


Image of die in horizontal position; slight wear in lower right quadrant.



Tilted die. Image shows reduction zone, parallel, and back relief. Note radial lines showing scoring.

The Profiloscope may also be used to measure the die angle by tilting the die, whereupon the outermost circle of light distorts to a heart-shaped image; by sufficient tilting, the cusp at the top of the heart may be made to touch the innermost circle. In this position the direct ray through the die and the outermost reflected ray coincide on the screen. By tilting the die in the other direction a similar situation will be seen, but on the other side. From the total angle of tilt of the platform, the die angle can easily be calculated.

A feature of interest is the ease with which the Profiloscope can be used for rapid quality control of die profiles and, in this connection, it is believed that the instrument will enable wire drawing works to plan batch replacements economically, thereby reducing this expenditure considerably.

The illustration shows the standard model Profiloscope, with a range of 4/0 SWG to 19 SWG. Die sizes below this range are covered by another instrument operating on the same principle.

NEW LIGHT-VESSEL BUILT IN BRITAIN FOR THE UNITED STATES

The most powerful sea navigation light of its kind in the world, with a maximum beam power of 5,560,000 international candles has been installed in the Ambrose Light-Vessel, which returned to its station 30 miles from New York, on May 29th, after re-commissioning by the United States Coast Guard.

The new apparatus was made by Chance Brothers Limited at Smethwick, Birmingham. The apparatus, which is the result of close collaboration between British and U.S. engineers, comprises a pendulum on double gimbal bearings, with the optical equipment situated above the bearing mounting. The purpose of this arrangement is to transmit the light beams from the optic sensibly parallel to the sea, to ensure continuous visibility to the mariner regardless of rolling or pitching of the light-vessel. Adjustment is provided within the weight box of the pendulum to cater for a relatively wide range of vessel period, i.e., 9 to 23 seconds, and allows for a maximum angle of roll of pitch of the vessel of 29 deg.

The optical equipment consists of four pairs of 18-in. × 6-in. focus mirrors arranged in tandem in a vertical plane, with a 1,000 W high-efficiency lamp at the focus of each mirror. Thus, a failure of a single lamp in any one pair of mirrors means only a temporary loss of candlepower of one aspect and does not destroy the identification of the flashing character exhibited by the light. A system of current-flow detection relays is employed in each pair of lamp circuits to indicate lamp filament failure.

The flashing character of the light is obtained by rotating the optical apparatus by means of an electric motor, and provision is made for a wide range of both flash characteristic and speed of rotation. Flexibility of flashing is made possible by means of selection of the number of aspects employed and by the angular displacement of the aspects relative to each other in the horizontal plane. Changes in speed of rotation of the optic are provided for by a selection of gear change wheels, the motor speed being maintained constant.

With the maximum beam candlepower emitted by this apparatus it would be possible for the light to be seen from a distance of 64 sea miles on a tropically clear night, if the earth's curvature permitted such a geographical range at sea level. The purpose of so high a beam candlepower, however, is not for clear weather conditions, but to enable the light to be visible to the mariner when atmospheric conditions are misty or even foggy. Assuming that the light would be visible at about 46½ sea miles in clear weather, with an atmospheric transmission as high as 0·85, at 0·60 the range would be reduced to 18½ sea miles and at 0·5 (representing a daylight visibility of six sea miles) the light would be seen at 14 sea miles.

To deal with such varying atmospheric conditions, the United States Coast Guard development engineers decided to incorporate a system of voltage control, by means of which the beam of candlepower can be selected and exhibited according to prevailing atmospheric conditions. Thus, at maximum operating voltage the beam power will be 5,560,000 candles; at 80 per cent voltage, 2,780,000 candles; at 65, 1,390,000 candles; and at 40, 278,000 candles. In practice, the lower beam candlepowers will be exhibited when the weather is clear or fairly clear and the maximum only when needed.

BANTAM SIZE AIR CLAMPS AND CONTROL VALVES

Schraders have introduced a range of bantam-size air clamps, for clamping and holding work for machining and processing. The Schrader air clamp consists of a push type spring return air cylinder designed to produce a short, sturdy stroke of the piston with sufficient power to hold the work firmly in place as long as required and with the same pressure every time.

The action of the air cylinder is rapid and the push power is smooth and constant. The return spring is fully enclosed and strong enough for rapid piston return, giving quick release action.

Two types, as illustrated, are available: Base mounted for power stroke perpendicular to the mounting surface, and leg mounted for power stroke parallel to mounting surface, respectively. Each is supplied with a one or two-inch stroke and in three cylinder diameters.

For work that has to be held in position for machining, these air clamps provide time-saving and effortless operation and reduce operator fatigue.

A series of two, three and four-way control valves has also been introduced for use with the Schrader air clamps. These valves incorporate the sliding disc principle. The valve body houses a precision-lapped stainless steel disc which slides on a bronze ring to provide a positive action air-tight seal. The valves are available for hand or mechanical operation. A typical hand-operated valve is shown above.

ON A 12-ROLL SUPER CALENDER

The illustration shows

a 12-roll super calender

built for export by

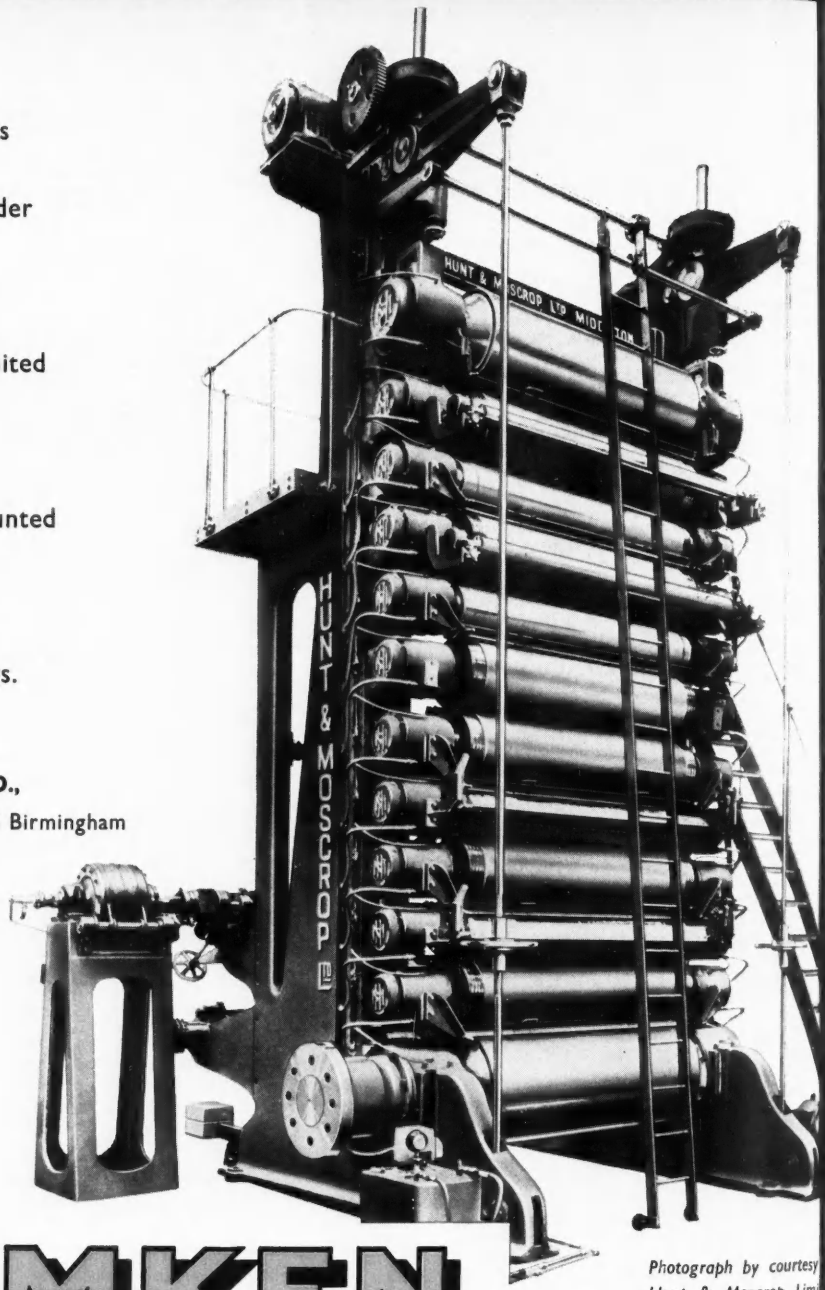
Hunt & Moscrop Limited

of Middleton.

All the rolls are mounted

on Timken

tapered-roller bearings.



Photograph by courtesy
Hunt & Moscrop Limi

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CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

BLOWERS AND COMPRESSORS

The design of Roots blowers

By H. ERTL. (From *Konstruktion, Germany*, Vol. 5, No. 3, March, 1953, pp. 71-77, 6 illustrations.)

ROOTS blowers can be designed to operate with a good overall efficiency for pressure differences up to 0·8 kg/cm². The overall efficiency is $\eta_{total} = \eta_{ad} \times \eta_{vol} \times \eta_m$, i.e., it is the product of the adiabatic, volumetric and mechanical efficiencies of the Roots blower.

To obtain basic data on leakage and frictional losses, 14 Roots blowers of types used for the scavenging of 2-stroke engines were tested on a rig with an electric motor drive. The rig losses were determined by previous calibration tests. The tests included 4 Deutz, 1 Spelleken-Deutz, 1 DVL, 1 Gräf & Stift, 1 Aerzener, 1 Argus, 2 Allison, and 1 Densham type blowers. In these tests, the total leakage cross-sections and the equivalent mean gaps were determined from experimental results. In addition, the following analysis of blower losses was developed :

Adiabatic power losses are calculated from the specific heat of the working fluid, the temperature difference and the effective air delivery. *Frictional* power can be assessed as the product of air viscosity, swept volume, speed squared, and a factor of 0·012 for L/D ratios around unity and speeds of 4000-8000 rpm. This factor is doubled or trebled for large L/D (rotor length/rotor diameter) ratios or with extreme throttling at the inlet or outlet. *Leakage* power depends on rotor shape, surface roughness, rotor and bearing clearances, and thermal expansion.

The experiments showed that the mean equivalent gap width $s_m = f_m/(3L + 3D)$, where f_m = total leakage cross-section, can be approximated by a straight line from 0·09 to 0·34 mm, for rotor diameters increasing from 35 to 275 mm. The *leakage flow coefficient* is $\alpha = Q_L/(wf_m)$, where Q_L = delivery loss and w = flow velocity in the gaps ; this coefficient increases as a function of the L/D ratio and the peripheral velocity. It was not possible to take account of throttling losses separately, despite their importance. Theoretical considerations based on these results show that the British trend to build " short and fat " Roots blowers is fundamentally correct.

The article includes a proposed method of calculation of required power and air delivery, a discussion of regulation by speed variation, by-passing, and throttling, and gives numerous references to previous literature.

DIELECTRICS

The Dielectric Losses of Various Filled Phenolic Resins

By R. L. LOFTNESS. (From a thesis presented to the *Swiss Federal Institute of Technology, Zurich*, 1952, 64 pages, 64 illustrations.)

THE dielectric losses of filled phenolic resins have been investigated and are reported at length.

A general discussion of the nature of dielectric loss and of the importance of ionic conductance, interfacial polarisation, and orientation polarisation is given. The dielectric loss theories of Debye and Wagner are presented. Simple formulae for the minimum dielectric loss of multi-component systems are stated.

A description of the experimental apparatus and procedures is given. The dielectric losses of filled phenol formaldehyde resins were measured primarily as a function of temperature, frequency, and type and quantity of filler ; and secondarily, as a function of the type of resin, moisture content, grain size of the filler, and extent of curing of the resin. Graphs are given illustrating the effect of the various variables on the dielectric loss. The fillers used were mica, quartz, talc, asbestos, aluminium oxide, glass, calcium silicate, titanium dioxide, iron oxide, corundum, wood flour, cotton, paper, and alphacellulose.

Resins containing the low-loss mineral fillers were found to have the lowest dielectric loss. The moisture-absorbing and ionizing characteristics of the filler were found to have a marked effect on the dielectric loss.

ELECTRIC CABLES

The Mechanical and Thermal Properties of Gas Compression Cables

By N. KLEIN. (From *Applied Scientific Research, Holland*, Section A, Vol. 3, No. 5, pp. 361-384, 10 illustrations.)

THE mechanical and rheological properties of gas compression cables subjected to inside and outside pressure are investigated with a specially made gauge and the thermal expension coefficient is measured. The mechanical and the thermal properties and the mode of pressure transmission through the cable make possible the description of the working mechanism of the cable. It is thought that the pressure drop in the lead sheath can be considerable when load is switched off the cable. The reinforcing tapes on the pressure transmitting sheath can become permanently extended. Fatigue failure of the pressure transmitting lead sheath is possible, due to the large strains. As the strains are smallest in the sheath of a circular cable, this is preferable to an oval one. Design considerations deal with the questions of polythene and lead alloy sheath, aluminium reinforcing tapes and pressure retaining tubes. It is also found that gas compression cables can be used for all voltages, including low ones.

GEARING

Tapered Spur Gears

By W. WOLFSTIEG. (From *VDI Zeitschrift, Germany*, Vol. 95, No. 6, February 21, 1953, pp. 173-174, 8 illustrations.)

SPUR gears are frequently required to mesh without any play between teeth. The fitting of cylindrical gears, or the mounting of shafts carrying these gears, is difficult and expensive under these conditions. If, however, the teeth are cut on a cone instead of a cylinder, then a simple axial displacement of one gear (Fig. 1), or one gear shaft (Fig. 2), will completely eliminate the play. Straight, as well as helical spur gears, can be tapered in this manner. The tapered gears are hobbed or generated with standard tools for cylindrical gears. In the hobbing process, the turntable carrying the work piece is given an additional motion away from the hob ; this motion is connected in a fixed ratio to the feed of the tool. If the gear is generated, the axes of cutter and work-piece are arranged to include the desired angle of taper. A taper between 1 : 10 and 1 : 30 is

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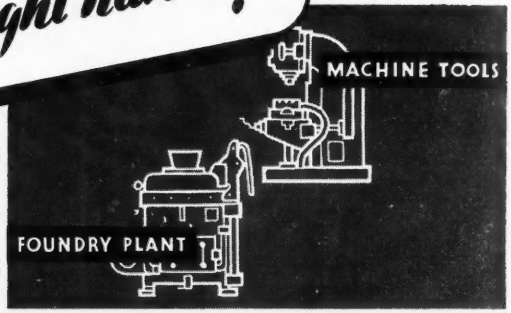

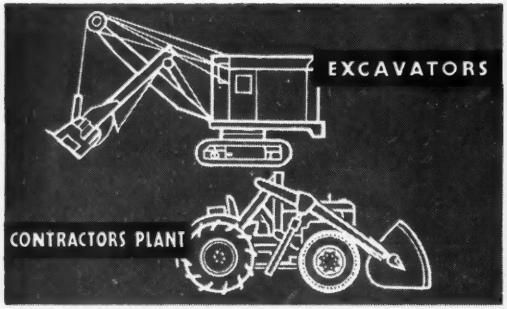

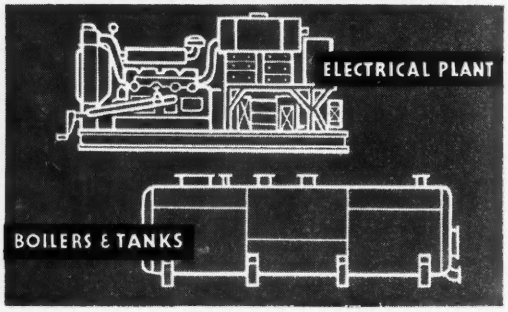

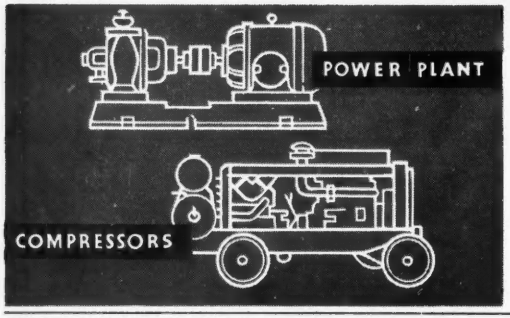

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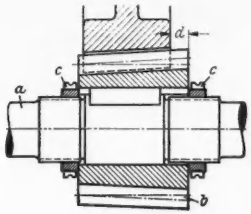


Fig. 1. **Tapered spur pinion axially adjustable on its shaft.** *a* = shaft, *b* = tapered spur pinion, *c* = nuts for axial adjustment, *d* = maximum axial displacement.

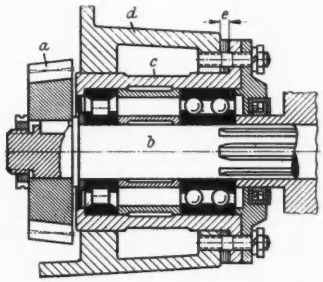


Fig. 2. **Tapered spur pinion on an axially adjustable shaft.** *a* = tapered spur pinion, *b* = shaft, *c* = bearing for shaft *b*, *d* = frame of machine, *e* = maximum axial displacement.

usually sufficient to eliminate the play caused by ordinary machining tolerances and wear of teeth. The correct meshing action of the tapered gear is obviously due to the fact that involute teeth are not bound to a definite pitch circle and can be modified, so that undercutting of teeth is avoided. The tapered involute tooth can be said to posses a modification varying linearly over its width ; the intersection of the pitch cone with the theoretical pitch circle need not be in the centre of the face width and may even lie outside the limits of the gear, in accordance with the usual methods of modifying the teeth of cylindrical gears.

METALLURGICAL ENGINEERING

Orienting Grains in Transformer Steel

By M. J. BOLTON. (From *General Electric Review*, *U.S.A.*, Vol. 56, No. 3, May 1953, pp. 13-16, 6 illustrations.)

THE grain-oriented silicon steel used in modern transformers has extremely uni-directional magnetic properties. Nearly 95 per cent of its grains are oriented in the rolling direction, and it has been developed to a point where its use as a core material means a reduction of more than 60 per cent in transformer size.

Insofar as grain orientation in the rolling direction is concerned, it may be said that a point of diminishing returns has been reached. The next logical step is, therefore, to alter the present orientation, i.e., to make the magnetic properties across the sheet, or at right angles to the direction of rolling, equally as good.

To obtain good magnetic properties in two directions, the mechanism of grain nucleation and subsequent growth must be understood. In other words, the method of making one grain grow, whilst suppressing the growth of others, must be known. One such method is by means of a series of closely controlled cold reductions, i.e., rolling at room temperature, followed by heat treatments. Under most conditions, well-oriented grains have a greater energy differential than poorly oriented grains, before and after heat treatments. To begin to understand this phenomenon, one approach is the determination of relative energies involved in boundary movements or grain growth.

Although the subject is still in its infancy, some interesting features have been observed, and experimental evidence has agreed fairly well with theoretical predictions. This lends encouragement to the belief that, from a practical viewpoint, the factors controlling the growth of grains in metals will one day be evaluated and understood.

The possibility exists that these energy studies will permit the growth of unwanted grains to be checked, and the growth of desired grains to be promoted. When this has been effected, the steel needed by engineers to reduce transformer sizes still further, will be produced.

POWDER METALLURGY

Machinability of Metal Powder Parts

By J. L. EVERHART. (From *Materials & Methods*, *U.S.A.*, Vol. 37, No. 4, April, 1953, pp. 98-100, 7 illustrations.)

ONE of the advantages of pressed and sintered metal powder parts is that machining is largely or completely eliminated in a great many cases. Consequently, less attention has been given to machinability than to other problems and less specific information is available. However, machining is required in the finishing of some metal powder parts where it is impossible to press the powder into the desired final shape.

Parts produced by powder metallurgy methods range in properties from low-strength relatively soft porous bodies to high-strength extremely hard carbides. The range of machining properties is also wide, but there are certain fundamentals inherent in the process which set these parts off from cast and wrought parts, even when the composition is similar.

By the nature of the process, all parts are affected to varying degrees by such factors as particle size of the powders, pressing practice, sintering conditions, composition, and density.

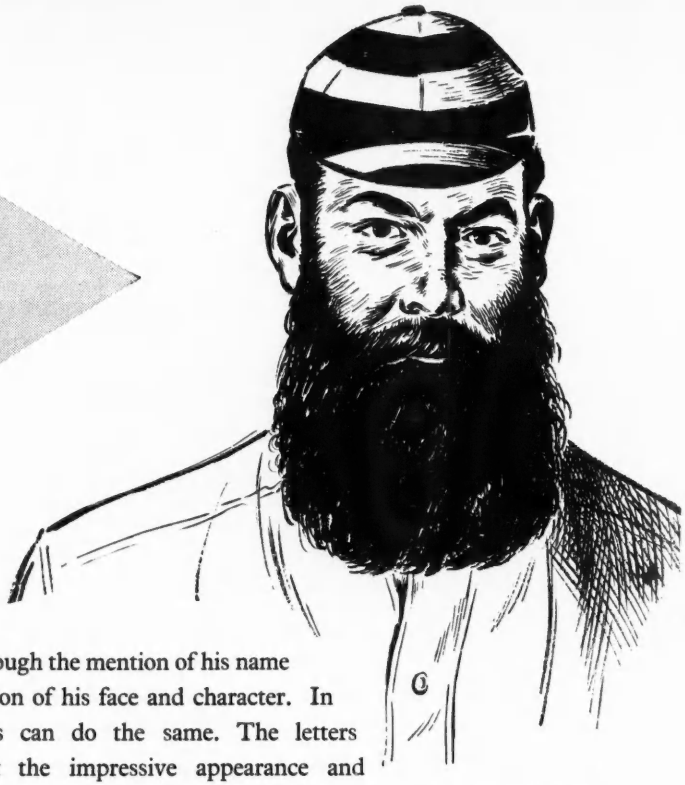
The combination of all these factors can cause variations in density and hardness in an individual part. These variations can make considerable difference in the relative machinability of two parts nominally of the same composition, and are probably responsible for the differences of opinion expressed by engineers in this connection.

The author outlines practical procedures for machining various types of parts. For instance, highly porous parts can be impregnated with molten salts to improve machinability. Sodium chloride is used to impregnate iron parts, but chlorides attack copper. Therefore, oxalic acid is used to impregnate copper-base compacts. After impregnation, the parts are machined with sharp carbide tools having top rake angles of 5 to 7 deg. side, and front rake angles of 5 deg. and no relief. Speeds of 500 surface feet per minute, depths of cut of 0·003 to 0·005 in., and feeds of 0·004 in. are satisfactory for finishing cuts. After machining, the salt is leached out with hot water. The porous condition of the part is retained by this procedure.

Also, it has been reported that the addition of 4 per cent molybdenum sulphide to iron with a porosity of 15 per cent resulted in a product which had machinability equivalent to yellow brass. At the present price of molybdenum sulphide, this method is not attractive as a commercial procedure, but it is possible that a less expensive sulphide might be substituted with similar results. Other parts have been impregnated with paraffin to improve the machinability, the paraffin being subsequently removed.

Indications are also given for the machining of dense metal powder parts and carbide parts.

THE POWER OF A NAME



WHEN a man is famous enough the mention of his name calls up an instant impression of his face and character. In certain cases mere initials can do the same. The letters 'W.G.' certainly suggest the impressive appearance and famous achievements of the most renowned cricketer of all time. Linked with the name Hoover, the initials 'F.H.P.' have a similar power. To manufacturers everywhere they represent a fractional horse power motor that is unsurpassed for efficiency and quality, that will give the same smooth service for the whole of its long life. These manufacturers have *proved* that the name Hoover F.H.P. represents the world's finest 'fractional' — with the world's finest service plan to back it.

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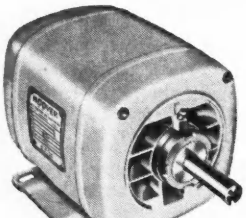

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WELDING

The Welding of Thick Plates of High-Strength Aluminium Alloys

By D. C. MARTIN, M. I. JACOBSON and C. B. VOLDRICH. (From *The Welding Journal, U.S.A.*, Vol. 32, No. 4, April, 1953, pp. 161-s-171-s, 9 illustrations.)

FOR the past five years, Battelle Memorial Institute has been conducting an investigation into the problem of welding thick plates of high-strength aluminium alloys.

Of the several high-strength aluminium alloys commercially available, three were selected for these investigations., viz. 61 S (containing 0·25 per cent Cu, 0·6 Si, 1·0 Mn, and 0·25 Cr); 24 S (4·5 Cu, 1·5 Mg); and 75 S (1·6 Cu, 2·5 Mg, 5·6 Zn, 0·3 Cr). Alloy 61 S was chosen because it was a widely used medium-strength structural alloy, and alloys 24 S and 75 S because they were widely used high-strength alloys.

For most of the work done during this investigation, the plates used were in the heat-treated condition that produced the highest strength. Three methods of welding heavy aluminium sections were used. These were:—

1. Metal-arc welding, using a flux-coated electrode.
2. Carbon-arc welding, using an uncoated filler-metal wire and a powder flux applied to the weld during welding.
3. Inert-gas-shielded tungsten-arc welding, using an uncoated filler-metal wire.

Gas welding was not used because it introduces a large amount of heat into the plate and produces a wide heat-affected zone and low-efficiency joints.

Results of these investigations over the past five years show that the welding of thick plates of high-strength aluminium alloys is somewhat less of a problem than it was previously. Advances in welding methods and techniques and the development of improved filler-metal alloys will enable the design engineer to use strengths considerably above those used in the past. In the cases where postweld heat treatments can be used, joint efficiencies of 100 per cent in 61 S, 80 to 90 in 24 S, and 50 to 60 in 75 S can be obtained with useful ductility. At present, the design of "as-welded" joints may be based on joint efficiencies of 70 to 80 per cent in 61 S, 60 to 70 per cent in 24 S, and 50 to 60 per cent in 75 S.

The obvious solution to the problem of obtaining high "as-welded" joint efficiencies is to use alloy base plates which are not affected by welding heat, e.g., aluminium-magnesium alloys. Although such alloys have found wide applications as structural material in Europe, their use in U.S.A. has been limited by lack of demand, because special techniques are required in their manufacture and because of their perhaps undeserved reputation for susceptibility to stress corrosion.

Another solution to the welding heat problem would be to use a process requiring no heat or, at most, temperatures lower than 300° F, i.e., cold welding. However, the process is now limited to relatively thin sections. It will be several years, if ever, before this process can be developed to the point where it can be used for thick plates of aluminium alloys.

The introduction of the inert-gas-shielded consumable-electrode process has been a tremendous step forward in the welding of aluminium. The porosity likely to be encountered in the weld metal is believed to be due to hydrogen present as water vapour or hydrated aluminium oxide on the surface of the filler-metal wire and base plates and to hydrogen occluded within the metals themselves. Careful attention to the cleanliness of filler-metal wire and base plates helps considerably in the reduction of porosity, as does the use of proper welding techniques. Perhaps refinements in aluminium manufacturing techniques will eventually eliminate the problem of weld-metal porosity.

SHRINKAGE OF PLASTICS DURING POLYMERISATION

(*Concluded from page 226*)

the preparation of the samples before insertion in the test apparatus. Under absolutely identical preparation conditions, these lines in Fig. 4 should always coincide. Diagrams of this 2nd type give, therefore, a possibility of verifying that absolutely identical conditions are obtained for tests.

SOURCES OF ERROR

By observing the precautions previously indicated, some of the possible sources of error, such as air bubbles in the mercury or in the test bag, time lags in the initiation of the test, and difficulties in obtaining temperature equilibrium at the beginning of the test, can be reduced or eliminated. However, errors may also be due to other causes, e.g., undesirable rises in temperature of the resin, owing to the exothermic character of the reaction and the poor heat conductivity of the resin.

The heat generated by the reaction of polyester type resins is considerable, particularly when large masses are employed. In a mass of resin of 1 litre, for instance, it was found that the temperature can reach 180° C. A study of the temperature conditions was therefore necessary. An iron-constantan thermocouple was inserted in the mass of resin in the rubber bag, and the glass stopper at the bottom of the apparatus was replaced by a rubber stopper sealed with picein. The test was carried out in the normal way, and measurements taken throughout the polymerisation process showed that the temperature deviation was not more than 0·1° C from the thermostat temperature.

Consequently, it may be said that, with samples of resin of about 1 gram, the reaction is maintained under isothermal conditions. As, in practical cases, the range of temperatures is never exactly the same, it is advisable to carry out tests at different thermostat temperatures. The investigation could be extended by using an automatic control system for the thermostat which would alter the water temperature in accordance with the variation occurring in a practical application.

Another source of error to be considered is the presence in solution of variable quantities of air or oxygen, which may be introduced during the preparation of the resin or the insertion of the catalysts and the hardening agents, although the specimen is kept under vacuum before these are added to it. In spite of this precaution, according to the age of the sample, the presence of peroxides may have an effect on the subsequent development of the reaction. When the catalysts and hardening agents have been introduced, it is no longer possible to create a vacuum, and the setting up of the apparatus must occupy as short a period of time as possible, to avoid further possibilities of error.

CONCLUSIONS

The proposed apparatus and test procedure should make it possible to determine: (i) The amount of shrinkage to be allowed for in the design of moulds; (ii) The conditions for minimum shrinkage with different types of fillers; and (iii) The most suitable type of catalyst and its optimum proportion for required conditions and mechanical properties. Furthermore, they should be useful in studying the kinetics of polymerisation under conditions as close as possible to those occurring in industrial applications.

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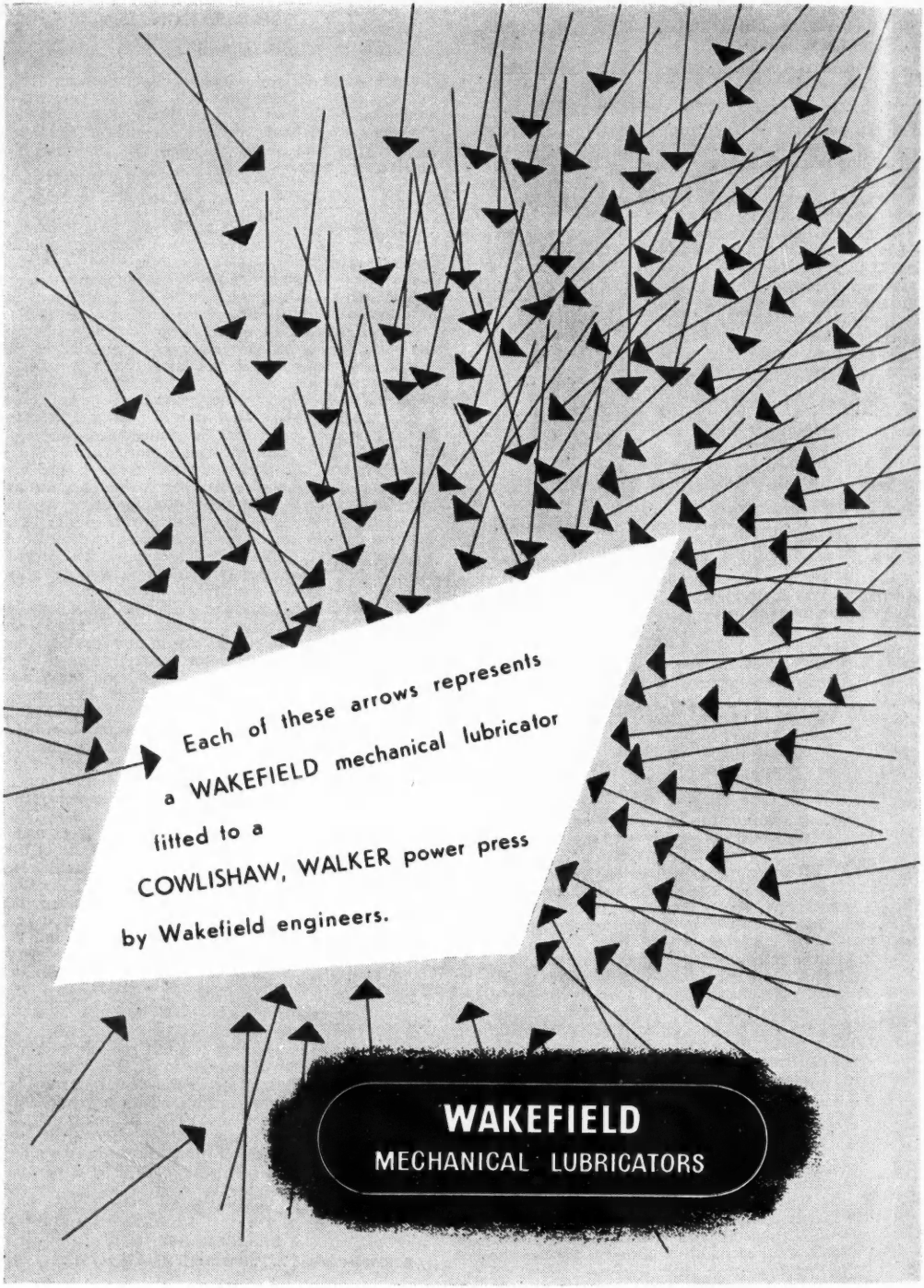
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NEWS OF THE MONTH

CORONATION HONOURS

The Coronation and Birthday Honours List of Her Majesty, The Queen, contains details of awards to technical men and industrialists associated with the engineering and allied industries. The prominent industrialists, Sir Peter Bennett and Sir Ralph Glyn, are in the list of baronies created, whilst Sir Percy Mills is included among the baronetcies. Some of the awards are listed below.

BARONS

Sir Peter Frederick Blaker Bennett, joint managing director of Joseph Lucas Ltd., Past-President of the Federation of British Industries, the Birmingham Chamber of Commerce and the Society of Motor Manufacturers and Traders.

Sir Ralph George Campbell Glyn, chairman of the Skefko Ball Bearing Co. Ltd. and a director of J. Samuel White & Co. Ltd., and other companies.

BARONETS

Sir Percy Herbert Mills, chairman of the National Research Development Council, vice-chairman and managing director of W. & T. Avery Ltd., a director of Textile Machinery Makers Ltd., and other companies, Past-President of the Birmingham Chamber of Commerce.

Major-General Sir Edward L. Spears, chairman of the Institute of Directors, chairman and managing director of Ashanti Goldfields Corporation Ltd., a director of Associated Portland Cement Manufacturers Ltd. and British Portland Cement Manufacturers Ltd.

Sir Herbert Geraint Williams, chairman of Bennis Combustion Ltd., a director of the Power Plant Co. Ltd., the Harper Automatic Machine Manufacturing Co. Ltd., S. Instone & Co. Ltd., and other companies.

KNIGHTS BACHELORS

Alderman Geoffrey Morris Barnet, director of John Jardine Ltd., Nottingham, Midland Machine Trust Ltd., Richard Simon & Sons Ltd., Ross Engineering Co. (Leicester) Ltd., and Sileby Engineering Co. Ltd.

Dr. Edward Crisp Bullard, F.R.S., director of the National Physical Laboratory.

Mr. Stuart Coldwell Goodwin, chairman and managing director of Goodwin & Co. Ltd., Neepsend, Sheffield, chairman and joint managing director of Neepsend Rolling Mills Ltd., and the Neepsend Steel & Tool Corporation Ltd., chairman of Jonas & Colver (Novo) Ltd. and G. Beaton & Son Ltd., director of Ferro-Alloys and Metals Ltd., Loxley Steel Works Ltd., Premier Steel Co. Ltd., Resisto Stainless Steel Co. Ltd., and other companies.

The Hon. Francis John Hopwood, chairman of the Shell Oil Co. Ltd. and managing director of the Shell Petroleum Co. Ltd. and its subsidiaries. He is also a director of the Elba Tinplate Co. Ltd., and other companies.

Major Sydney Frank Markham, M.P., director of Richard Johnson, Clapham & Morris Ltd., Trafford Park, Manchester.

Mr. Walter James Drummond, M.I.C.E., M.I.Mech.E., deputy chairman of the National Coal Board.

Mr. Stanley Walter Rawson, director-general of machine tools at the Ministry of Supply. Mr. Rawson is a director of the Clydebank Shipbuilding & Engineering Co. Ltd. ; the Cravens Railway Carriage & Wagon Co. Ltd. ; Cravens Ltd. ; Firth Brown Engineers Ltd. ; Firth Brown Steels Ltd. ; Hard Metal Tools Ltd. ; High Speed Steel Alloys Ltd. ; International Shipbuilding & Engineering Co. Ltd. ; Light Machines Ltd. ; Michell Bearings Ltd. ; Nitralloy Ltd. ; Powderloys Ltd. ; Tool Metal Manufacturing Co. Ltd., and other companies.

Mr. Harold Charles West Roberts, B.Sc., M.C., C.B.E., Chief Inspector of Mines and Quarries, Ministry of Fuel and Power.

Mr. William Reginald Vernon Smith, joint managing director of the Bristol Aeroplane Co. Ltd., director of Babcock & Wilcox Ltd., Rotol Ltd., and other companies.

Mr. Thomas Octave Murdoch Sopwith, F.R.Ae.S., chairman of the Hawker Siddeley Group, joint managing director of the Gloster Aircraft Co. Ltd., A. V. Roe Ltd., and other companies.

K.C.B.

Sir John Douglas Cockcroft, F.R.S., F.R.Ae.S., chairman of the Defence Research Policy Committee, scientific adviser to the Minister of Defence, director of the Atomic Energy Research Establishment.

K.B.E.

Mr. John Anthony Carroll, deputy controller (Research and Development) Admiralty.

Sir Greville Simpson Maginnes, chairman and managing director of the Churchill Machine Tool Co. Ltd., deputy chairman of Tube Investments Ltd., chairman of Associated British Machine Tool Makers Ltd.

Sir Frederick Ernest Rebeck, chairman and managing director of Harland & Wolff Ltd., Belfast. He is also a director of Colvilles Ltd., Short Bros. & Harland Ltd., and other companies. Sir Frederick is vice-president of the Institution of Naval Architects, past-president of the Institute of Marine Engineers and of the Shipbuilding Employers' Federation, joint chairman of the Technical Committee of Lloyd's Register of Shipping and a Belfast Harbour Commissioner.

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At the end of 1950 the Admiralty appointed a committee called the Admiralty Oil Quality Committee firstly to review questions of the quality of fuels and lubricants for the Royal Navy and secondly to advise whether a permanent Committee should be established to keep such matters continuously under review.

Upon the recommendation that such a permanent Committee should be formed, the Admiralty has set up the Admiralty Fuels and Lubricants Advisory Committee under the Chairmanship of the Hon. R. C. Geddes. The Committee consists of :—
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Dr. C. M. Cawley, O.B.E., Department of Scientific and Industrial Research,
Mr. H. S. Gibson, C.B.E., President, Institute of Petroleum,
Mr. I. G. Barr-Adams, Past President, National Lubricating Oil & Grease Federation,
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Mr. C. Lawrie, Vacuum Oil Co.,
Mr. C. H. W. Smith, Anglo Iranian Oil Co.,
Mr. D. B. Vale, The Shell Petroleum Co.

Representatives from the Ministry of Fuel and Power and from various Admiralty Departments also serve on the Committee.

The Committee has appointed a Fuels Technical Sub-Committee under the Chairmanship of Professor F. H. Garner, O.B.E., and a Lubricants Technical Sub-Committee under the Chairmanship of Dr. C. M. Cawley, O.B.E.

Apart from members of the Main Committee serving on these Sub-Committees, the following are also members :—

Fuels Sub-Committee
Mr. A. E. Gerty, Regent Oil Co.,
Mr. C. W. G. Martin, The Shell Petroleum Co.,
Mr. F. G. Richards, Vacuum Oil Co.,

Lubricants Sub-Committee
Mr. R. W. L. Fox, Vacuum Oil Co.,
Mr. N. G. Gullick, Regent Oil Co.,
Mr. A. R. Stark, Anglo Iranian Oil Co.,
Mr. R. S. Teale, The Shell Petroleum Co.,
Mr. G. H. Thornley, C. C. Wakefield & Co.,
Mr. E. H. Wild, Esso Petroleum Co.,
Mr. J. M. Wilson, Anglo Iranian Co.

PERSONAL

Mr. John Alcock, chairman and managing director of the Hunslet Engine Co. Ltd., Leeds 10, has been elected President of the Locomotive Manufacturers' Association.

Mr. P. T. Bliss, M.I.S.M.A., has been appointed sales director of G. A. Harvey & Co. (London) Ltd., Greenwich Metal Works, London, S.E.7.

Mr. G. V. Bloomfield, B.Sc., A.Inst.P., has been appointed senior development engineer of the Hymatic Engineering Co. Ltd., in succession to **Mr. R. B. Elliott,** who has joined the English Electric Co. Ltd.

Lieut.-Col. T. Child, T.D., B.Sc., M.I.Mech.E., has been elected to the Board of the British Engineers Small Tools and Equipment Co. Ltd., Buckingham House, Buckingham Street, London, W.C.2.

Mr. R. E. Dawtrey has been appointed chief engineer of Sheepbridge Engineering Ltd., Chesterfield, and will be responsible for all metallurgical, chemical and engineering research and development.

Mr. Harold Flavell, chief buyer of Herbert Morris Ltd., Loughborough, has been elected a director of the company.

Mr. R. K. Fleming is relinquishing his position as managing director of the Igranic Electric Co. Ltd., Bedford, on June 30, and has been elected chairman of the company in succession to **Mr. G. R. Thursfield.**

Mr. A. F. C. Gardner has been appointed nothern general manager of Gibbons Brothers Ltd., Dudley, Worcestershire, with headquarters at " Cranbourne," Egglescliffe, Nr. Stockton-on-Tees, Co. Durham.

Mr. G. W. H. Gardner, C.B.E., has been appointed director general of technical development (Air), Ministry of Supply, in succession to **Mr. S. Scott Hall, C.B.**

Mr. T. F. Hargreaves, B.Sc.(Eng.)., A.M.I.E.E., has recently joined the Equipment Division of Mullard Limited as a consultant on the overall planning of radio and telephone communication systems and to consider general applications of electronics.

Prior to joining Mullard, Mr. Hargreaves was with Standard Telephones and Cables Limited, where he was engaged on multi-channel carrier development and the design of carrier telephony circuits.

Mr. T. G. Hicks, M.A.(Cantab.), has been appointed joint managing director of Power Jets (Research and Development) Ltd.

Mr. J. W. Hislop has joined Lang Pneumatic Ltd., Wolverhampton, and will operate from the London office in conjunction with **Mr. L. A. Boobyer,** London sales manager of the Company.

Mr. H. E. Jones, former secretary of the British Engineers' Association, has been appointed secretary of the British Association of Machine Tool Merchants, West India House, 96/98 Leadenhall Street, London, E.C.3.

Mr. Alonzo Limb, managing director of C. C. Wakefield & Co. Ltd., 46 Grosvenor Street, London, W.1, has been elected to the additional position of deputy chairman. **Mr. L. W. Farrow, C.B.E.,** is chairman of the company.

Mr. R. W. M. MacAlpine, A.M.I.E.E., Assoc.Am.I.E.E., has been appointed assistant manager of the British Thomson-Houston Co. Ltd., London district office, Crown House, Aldwych, London, W.C.2.

Mr. J. C. M. MacLagan, managing director of Greenwood & Batley Ltd., Leeds, has been appointed part-time member of the Yorkshire Electricity Board.

Captain Reginald Trevor Paul, C.B.E., R.N. (ret'd.), A.M.I.E.E., commercial manager of the Equipment Division of Mullard Ltd., has been appointed a director of Telcon Telecommunications Limited, in place of **Dr. D. B. Foster,** who has resigned from the Board owing to pressure of work in other directions.

Mr. Sydney Player has resigned his position as chairman of M.T.E. Control Gear Ltd., Leigh-on-Sea. He will retain, however, his position as chairman of the Newall Group of Companies.

Mr. G. Alan Pochin has been appointed deputy chairman, and **Mr. James Gavin** deputy managing director of Goodwin Barsby & Co. Ltd., St. Margaret's Ironworks, Leicester.

Mr. Walter C. Puckey, M.I.Prod.E., F.I.I.A., has been elected President of the Institution of Production Engineers. He has recently retired from the Ministry of Supply where, as Deputy Controller of Supplies (Aircraft Production) he had for the past 27 months an appointment of greatest importance. Previous this appointment, Mr. Puckey was a director and general works manager of Hoover Ltd., and he is also a director of British Tabulating Machine Co. Ltd. **Mr. Harold Burke, M.I.Mech.E., M.I.Prod.E., M.I.I.A.,** has been re-elected chairman of the Council. Mr. Burke is joint managing director of Concentric Manufacturing Co. Ltd., Birmingham and is also a director of Metaducts Ltd., Brentford, Rowmill Metals Ltd. and T. Miller Ltd., Birmingham. **Mr. G. Ronald Pryor, M.I.Prod.E.,** managing director of Edward Pryor & Son Ltd., Sheffield, has been re-elected vice-chairman of the Council.

Sir Eric Rideal, M.B.E., D.Sc., F.R.S., Professor of physical chemistry at King's College, Strand, London, W.C.2, has been appointed chairman of the Minister of Supply's advisory council on scientific research and technical development. He succeeds **Professor Sir John Lennard-Jones, K.B.E., Sc.D., F.R.S.,** who has been recently appointed Principal of the University College of North Staffordshire.

Mr. W. A. Rigby has been appointed northern divisional sales manager of the Industrial Products Department of Hoover Ltd., Hoover House, Regent Street, London, W.1, with headquarters in Manchester. **Mr. A. R. H. Barton** has been appointed southern divisional sales manager with headquarters at Perivale, Middlesex.

Mr. Clement Stephenson, general manager of the shiphuilding department of William Doxford and Sons Ltd., has been appointed a director of the company.

Mr. C. J. Teasdel has resigned his position with Crompton Parkinson Ltd., and has joined the Board of M.T.E. Control Gear Ltd., Leigh-on-Sea, Essex, as sales director.

Mr. W. E. Thompson, secretary of Laycock Engineering Ltd., Archer Road, Sheffield, has been appointed a director of the company. **Mr. Kenneth Walker, A.M.I.Mech.E., A.M.I.P.E.,** has been appointed works director.

BUSINESS NOTES

The Electrical and Radio Federation (Victoria), Dominion Chambers, 59 William Street, Melbourne C.1, Victoria, announces that a large Electrical and Radio Exposition will be held in October, 1954, in the Exhibition Building, Melbourne. Exhibits from all parts of the world will consist of electrical appliances and equipment, radio, radar and telecommunication equipment.

Vokes Ltd., Henley Park, Guildford, has arranged a special feature exhibition at the Imperial Hotel, Birmingham, which will be open until Saturday, June 20th. A representative selection of their complete range of filtration and silencing equipment will be shown.

Tecalemit Limited announces the establishment of Regional Sales offices each under the control of a Regional Manager, who will be responsible for the development of Sales of the Company's automotive and industrial products and the conduct of its affairs within the region. This arrangement will enable the Company to give improved service to its customers.

SCOTTISH REGION (also covering NORTHERN IRELAND)
Regional Office: 86, Cambridge Street, Glasgow, C.2.
Regional Manager: Mr. E. R. Hart.
Assistant Manager: Mr. D. A. Monaghan.
Industrial Supervisor: Mr. J. J. Bell.

NORTHERN REGION (also covering ISLE OF MAN)
Regional Office: North Road, Wetherby, Yorks.
Regional Manager: Mr. S. H. Smith.
Industrial Supervisor: Mr. J. G. Maxfield.
Motor Trade Supervisor: Mr. S. C. Kennett.

LONDON & MIDLAND REGION
Regional Office: Gt. West Road, Brentford, Mddx.
Regional Manager: Mr. L. R. Hedgcock.
Industrial Supervisor: Mr. W. Donaldson.
Motor Trade Supervisor: Mr. G. W. Ely.

The WESTERN & SOUTHERN REGIONAL OFFICE is located at the Company's main office at Plymouth, Devon.

Short Bros. & Harland Ltd. are opening a London Design Office. This will be in addition to their London Headquarters at 17 Grosvenor Street, W.1, and will be in Ozonair House, Longmoore Street, Victoria, S.W.1.

New Cement Factory in Hungary. Work has begun on the construction of a giant new cement factory on a 370-acre site near Vác, on the northern bend of the Danube in Hungary.

This factory will be considerably larger than the almost completed one at Hejöcsaba, whose fully-mechanised equipment will be turning out 1,000 tons a day by the end of this year. Output of the Danube Cement Works, as it is to be called, will be double that of any existing factory in Hungary. Its main building—360 yards by 65 yards—will house two rotary kilns, each 165 yards long, for production of clinker.

Existing cement factories are being extended and modernised. The big works at Lábatlan has been fitted with completely new plant, and new machines have been fitted at the Bélapátfalva factory.

PAREX Centres

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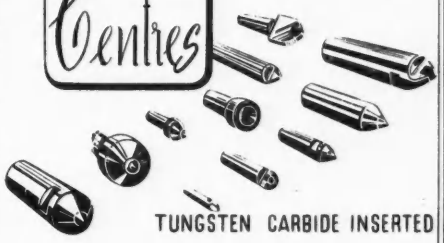

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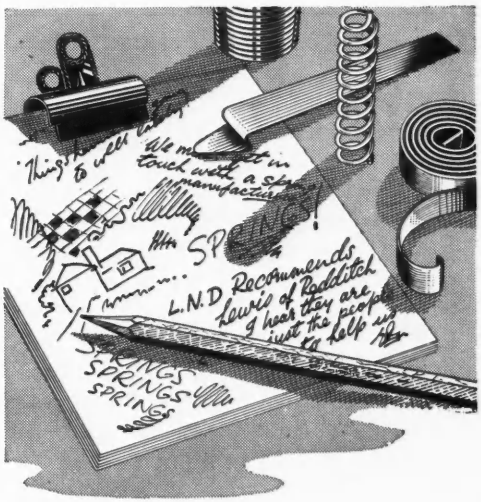
Tel. Tra. Park 1381.



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AN INTERESTING COMPRESSOR DRIVE

A synchronous induction motor rated at 2,200 hp at 1,500 rpm has recently been built by The General Electric Co. Ltd., for African Explosives and Chemical Industries Ltd., North Rand. The motor will drive a rotary compressor in the nitric acid plant at a dynamite factory, and an interesting feature of the drive is that the motor will operate in tandem with a gas turbine designed to burn a fuel which is a by-product of the chemical plant. The turbine will normally assist in driving the compressor, though it will at times be necessary to operate the plant for long periods with the motor alone. When it is operating in conjunction with the gas turbine the output of the motor is 670 hp ; the full output of 2,200 hp is required for starting and for running the plant when the turbine is not in use. Both the compressor and turbine are being built by the Escher Wyss Engineering Works Ltd., of Zurich.

The driving motor is a 6·6 kV, 3 phase machine of the screen-protected type, and is designed to operate at unity power factor at an altitude of 5,400 ft, the maximum ambient temperature being 40° C. Axial ventilation is employed, and the motor is fitted with pedestal journal bearings which are arranged for both oil-ring and forced lubrication. Excitation is provided by direct-coupled main and pilot exciters, and the motor is coupled to the compressor through speed-increasing gears with a ratio of 1500/8000 rpm

ELECTRICITY AND NATIONAL PROSPERITY

" Our inherent skill and craftsmanship must be made to yield increased dividends from world markets. Britain can make it, and no one must be able to make it better." This call and challenge to British industry was made at Torquay by Col. B. H. Leeson, C.B.E., T.D., M.I.E.E., Director, The British Electrical and Allied Manufacturers' Association and President of the Institution of Electrical Engineers. He was speaking on " Electricity and National Prosperity " to the fifth British Electrical Power Convention.

Colonel Leeson said the shortage of coal was a principal cause of our economic instability but of no less importance was the decline in the intensity of human endeavour. The challenge that faced the nation today was still one of re-orientating our productive effort so that it might take account of the basic changes in world economy and dovetail into the world's demands. There was no reason why Britain with her natural resources of coal and the human qualities of discovery, inventiveness and craftsmanship could not, if she was prepared to make the necessary effort, win economic recovery and prosperity in the " New Elizabethan Era."

Steps must be taken to pass from the surgery of import cuts to the needed cure of freeing and creating—not handicapping—greater world-wide trade. The research, development and state of the electrical industry was such that it was ready to make a further contribution in re-equipping industries and improving their productivity and hence their ability to export, like the electrical industry itself, against growing competition the world over.

The pattern of British overseas trade was changing. Once we were the workshop of the world, manufacturing goods of varying kinds to suit our external markets. But today the newly established light industries abroad were able to sell many of their products on their home markets at considerably lower prices than Britain. No longer could Britain expect to supply cheap cotton goods, for example, to India and the Far East to the same extent as a generation ago. We must concentrate our efforts on producing goods which would be unsurpassed in quality and bought for that consideration rather than price. Our inherent skill and craftsmanship must be made to yield increased dividends from world markets. " Britain can make it, and no one must be able to make it better."

Next to the land, Britain's coal deposits were the most important primary asset. The most modern processes and techniques of industry demanded ever-increasing quantities of heat, light and power. Industrial productivity was largely dependent on the amount of energy available to the workpeople. Coal and electricity were the parents of prosperity. The coal supply, however, was falling far behind the potential demand. To conserve our coal the ideal would be not to export it, but this was impracticable.

Col. Leeson said that if we were to increase productivity, the prime requirement was a plentiful supply of power. The problem which faced them was whether capital allocation for supply equipment would be sufficient to take into account rapidly increasing applications of electricity ; the replacing of obsolete and obsolescent plant to obtain fuel economy, and the need of industry for increased electrical horse-power to achieve greater productivity. The early attainment of a yearly rate of plant installation, reaching 2,000 megowatts, was imperative, and a capital investment policy allowing this rate of installation was a necessity to economic recovery.

From a recorded value in 1907 of £14 million the output of the electrical manufacturing industry has risen to an estimated total of £730 million in 1952 ; the industry now provided employment for nearly 600,000 people. It was the second in the list of exporting industries with £218 million.

One of the fundamental requirements of the expansion of exports was that the industry concerned should have a stable, progressive and resourceful home market—" a base from which the merchant adventurer in the new Elizabethan Era may set forth with both the enterprise and the wherewithal to undertake the venture and the risks that are inevitably involved."

To this great end, Industry looked to the Government to implement as soon as it can its undertaking in relation to export expansion that was given in the 1952 Economic Survey, namely,

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"to create and maintain economic conditions favourable to such expansion." For example, home markets must be freed from all artificial handicaps, such as purchase tax and other restrictive controls, so that the manufacturer might have the necessary quantity production and other facilities that would place him in a position to compete with the world.

The task of British industries in expanding their exports lay increasingly in their ability to supply products having the quality and the performance that could not be purchased elsewhere. Plants flourished best from roots in fertile soil; exports from growing home demands. Another issue was the expedient for restoring our balance of trade by import cuts. More freedom in world trade was essential to further expansion of exports.

The nation's economic system was struggling, with insufficient reserves, to maintain a precarious balance between what was put into it and what was demanded from it.

"Are people prepared to have less, or contribute more?" asked Col. Leeson. "We face to-day, a challenge and a choice. Are we as individuals and as a nation content to accept with complacency the obvious trend of affairs towards lower standards of living—or, alternatively, do we not take up the challenge this represents, and by courage and the exertion of more strenuous individual efforts on things that really matter change the trend towards recovery and better times for one and all? This is a human problem to which there can be only one answer—the challenge must be met before it is too late."

The general lines on which this challenge should be taken up was equally clear in principle although, putting them into practice might involve many problems. Those lines could be expressed in the form of four basic needs in the interests of prosperity:—

(a) More human effort; that is, more man-hours of work.
(b) Greater production of the basic commodities such as coal, metals, chemicals and other essential materials.
(c) More fuel and power for industry and the home.
(d) More capital; that is, greater economy and savings by Government, industry and individuals.

The answer to the challenge to-day of retaining our standard of living and improving our economic stability was the need for greater and more united human efforts on things that really mattered to obtain greater output and trade from the country's available resources. This must be accomplished first in the primary industries, such as coal, in order to increase their output and trade. As their output and trade increased, so the beneficial effects would extend to subsidiary industries and ultimately spread to all industries and activities.

Col. Leeson concluded "The problem is how to arouse public appreciation of the country's dire need. Much has been done, but a clarion call to John Bull to take off his coat and exert further efforts towards prosperity and peace is imperative. After all, national prosperity stems from its people, from its homes, and from its industries; if every individual will strive in order to thrive himself, then the nation's prosperity will look after itself."

BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 24, Victoria Street, London, S.W.1.)

Copper for Electrical Purposes. Tubes (High Conductivity). B.S. 1977 : 1953 is the fourth in a series of standards for copper for electrical purposes. The others are B.S. 1432, B.S. 1433, and B.S. 1434; sheet and strip, bar and rod, and commutator bars, respectively.

The standard lays down requirements of copper tubes of high conductivity and does not apply to cable sockets or conduit for electrical wiring for which the appropriate standards are B.S. 91, "Electric cable soldering sockets," and B.S. 840, "Light-gauge seamless copper and copper-alloy conduit and fittings for electrical wiring."

Provision is made whereby the purchaser can by special arrangement specify that tubes shall be made from oxygen free high conductivity copper complying with B.S. 1861 recently published. Price 2/6.

Classified Advertisements.

The rate for all classified advertisements is 6d. per word; in bold print 9d. per word; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

SITUATIONS VACANT

ARMSTRONG SIDDELEY MOTORS CAR DIVISION
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ARMSTRONG SIDDELEY require a first class Mechanical Designer aged 26/32 for their Project Design Office. Applicants must be works trained designers with at least Higher National Certificate and a sound background experience of aero engine design. The position offered is one of great interest and is an exceptional opportunity for a young engine designer to expand his experience in the gas turbine industry. Reply in detail to Reference HSR.3, Personnel Manager, Armstrong Siddeley Motors, Coventry.

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NEW ALL STEEL PRESS BRAKE for sale. Capacity 8ft. × 1/8 in. Between frames 69 in. Stroke 3 in. Die space 8¾ in. Completely motorised 400/3/50. Weight about 9 tons. **F. J. Edwards Ltd.,** 359, Euston Road, London, N.W. Tel.: Euston 4681.

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TRANSLATIONS (Technical, Commercial), all languages. Abstracts also supplied. **Olympia Translation Service,** 149 Blythe Road, London, W.14. RIVerside 5135.

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LATEST INDUSTRIAL LITERATURE

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Included in the brochure are details of design considerations for diecastings, finishing processes, methods of soldering and welding, and photographs of various diecastings made with this product.

2. Resilient-Grip Mounting Bolt. A newly-developed shock and vibration proof mounting bolt is described in a 6-page leaflet. Rubber is used as the gripping medium and the bolt is suitable for concrete, masonry, brick, wood, steel, and similar materials.

The bolt has, in addition, shock-absorbing qualities which can be enhanced by using it in conjunction with vibration-reducing materials, such as rubber matting.

3. Synthetic Resin Core-Binder. Booklet describes a synthetic resin core-binder and its method of use, and lists its advantages over conventional oil-binders.

4. Chill Cast Phosphor Bronze. A useful 8-page " weights and measures " booklet gives specifications of chill cast phosphor bronze solid and cored bars, with outside diameters ranging between $\frac{1}{2}$ and $10\frac{1}{2}$ in.

5. Variable Speed Unit. Lavishly illustrated brochure describes the salient points of construction and principle of operation of a variable speed unit featuring finger-tip control from 500 to 4,500 rpm.

Models ranging from $\frac{1}{4}$ to 14 bhp are available and any special or unusual speed variations can be catered for by the addition of a special reduction gear mounted on the output shaft of the unit.

6. Technical Survey of Copper. An interesting survey of copper and its alloys is contained in a 40-page publication. Contents include information on foundry practice, joining, physical and powder metallurgy, corrosion, testing, applications, and properties (including aluminium bronze, beryllium copper, and iron-bearing cupro-nickels).

The survey also contains a list of similar free technical publications covering a wide field of applications of copper and its alloys.

7. Instrument Mountings. Leaflet gives description and specifications of a wide range of anti-vibration mountings specially designed for scientific instruments and light machinery. The mounting consists basically of rubber, chemically bonded to metal, the rubber acting as the anti-vibration medium. It can be loaded in any plane and in any combination of torsion, shear, compression, or tension.

8. Wrought Aluminium Alloys. 20-page booklet provides, in convenient tabular form, technical data on a large range of wrought aluminium alloys, including composition, classification, weights, strength, applications, etc.

The information contained in this well-presented booklet should enable designers and engineers to select alloys most suited to their requirements.

9. Worm Reduction Gears. 12-page booklet describes and lists a range of fan-cooled worm reduction gears. The tooth form of the gearing is of special design to ensure silent operation and a feature of the unit is a large diameter quick-release top cover, for rapid inspection and replenishment of oil.

These gears are standardised in a series of progressive sizes, and outline drawings and tables of main dimensions are given in the booklet.

10. Vapour Blasting Equipment. An excellently produced and comprehensive 22-page booklet describes an abrasive vapour blasting process and gives details of its scope and applications. Drawings and photographs of various units are included.

11. High-Pressure Hose Units. Details are given in two 4-page leaflets of a wide range of high-pressure flexible hose units, many of which feature the new completely detachable end fittings. These units are of proven value in many types of installation, from aircraft engines to machine tools.

12. Injection Moulding Machine. In a 4-page illustrated leaflet the constructional features and specifications of a new 4 or 5 ounce injection moulding machine are given in detail. The machine is designed for manual, semi-automatic, or fully automatic operation, as desired by the operator.

A supplementary leaflet gives details of a new mould temperature control unit for preheating and maintaining accurate temperatures. Water is employed as a heat transfer medium and the unit is capable of operating to 220° F without any loss by evaporation, by the use of a closed, pressurised system.

13. Gas Cleaning Plant. A 6-page booklet describes a new design of cleaning plant which, in addition to giving a very clean gas with an impurity content not exceeding 0·5 grains per 100 cu ft, recovers the by-products as water-free pitch and oil.

The booklet includes a sectional drawing of the plant.

14. Precision Cylindrical Grinder. An illustrated leaflet and insert describes a precision cylindrical grinding machine and a new internal grinding attachment, thus enabling the machine to be used for both internal and external grinding to very fine limits.

15. Universal Oil-Grooving Machine. Well-illustrated brochure gives details of a new machine designed to produce multi-oil grooves without individual indexing.

Conventional methods of grooving may be satisfactory for meeting average demands but fall short when large batch production is required. The new machine cuts multi-grooves singly or continuously and can also be used for producing eccentric face grooves, " broken " grooves, taper grooves and straight grooves or splines.

From one to twenty-four grooves per piece can be cut, generation being by means of compound " pick-off " gearing.

16. Rod Mills. A leaflet gives basic technical data of end peripheral discharge, centre peripheral discharge, and open end rod mills, used in mining and quarrying for the fine grinding of wet or dry ores.

It is interesting to note that these rod mills were previously available in U.K. as a dollar importation, but are now available for Sterling.

17. Greasing Appliances. An illustrated 16-page catalogue gives specifications of numerous greasing appliances, including grease guns and nipples, tubing and fittings, and industrial greasing equipment.

18. Engineering Components of Nylon. The use of injection-moulded nylon in the manufacture of precision engineering components is increasing in popularity. A leaflet details some of the engineering applications of injection-moulded nylon. Separate leaflets give reprints of articles dealing with the use of nylon for ball bearings, and for bearings and gears respectively.

19. Surface Heating Equipment. In a 16-page, profusely illustrated booklet, many applications of an interesting method of electric surface heating are described. The heating medium is flexible, so as to conform to the surface, and may be had in the form of mantles, panels, jackets, and even tapes.

This method of heating has a wide range of applications, from chemical industries to glass plant works and aircraft factories.

20. Small Zinc Alloy Castings. Numerous examples of small zinc alloy die castings are illustrated in a brochure from a firm specialising in the production of such castings weighing under 0·5 ounces. Fully automatic machines are employed, capable of producing up to 225 pieces per minute.

Included with the booklet are separate reprints of articles dealing with small zinc die castings and high-speed die castings respectively.

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